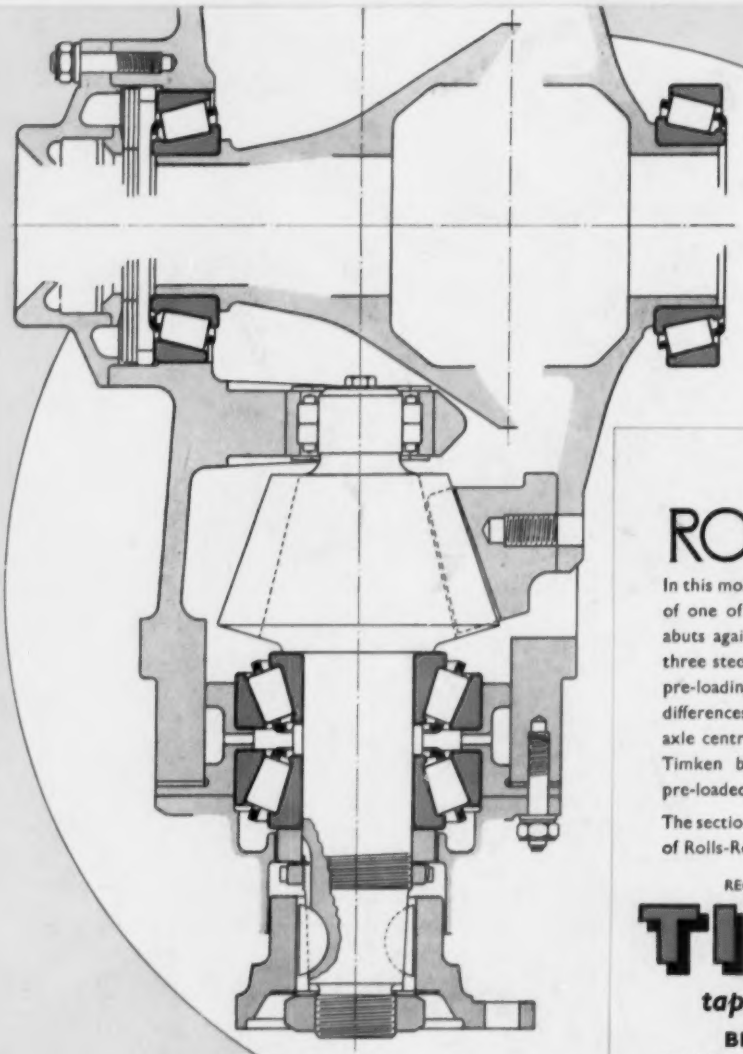
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 46 No. 10 **OCTOBER 1956** PRICE: 3s. 6d.



ON THE
ROLLS-ROYCE

In this mounting of the differential cage, the cup of one of the Timken tapered-roller bearings abuts against a pre-loaded spring consisting of three steel discs. This refinement maintains the pre-loading notwithstanding the expansion differences between the high-tensile light-alloy axle centre and the differential assembly. The Timken bearings for the driving pinion are pre-loaded by the ring nut.

The sectional drawing is reproduced by courtesy of Rolls-Royce Limited.

REGISTERED TRADE MARK: TIMKEN

TIMKEN

tapered-roller bearings

BRITISH TIMKEN LTD.
DUSTON, NORTHAMPTON (HEAD OFFICE);
& BIRMINGHAM

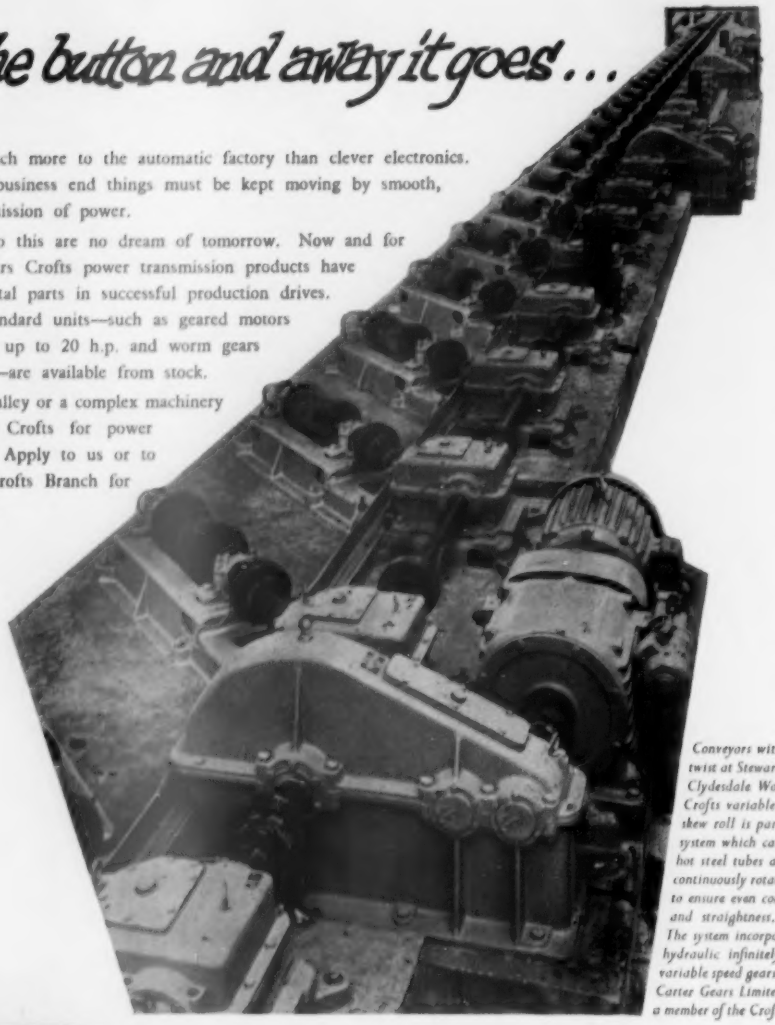
MADE IN ENGLAND

Push the button and away it goes...

but there's much more to the automatic factory than clever electronics. Down at the business end things must be kept moving by smooth, faultless transmission of power.

Machines to do this are no dream of tomorrow. Now and for many busy years Crofts power transmission products have played their vital parts in successful production drives. Better still, standard units—such as geared motors from fractional up to 20 h.p. and worm gears up to 65 h.p.—are available from stock.

For a simple pulley or a complex machinery drive, call in Crofts for power transmission. Apply to us or to your nearest Crofts Branch for further details.



Conveyors with a new twist at Stewarts & Lloyds' Clydesdale Works—this Crofts variable speed skew roll is part of a system which carries hot steel tubes and continuously rotates them to ensure even cooling and straightness. The system incorporates hydraulic infinitely variable speed gears by Carter Gears Limited, a member of the Crofts group

Crofts

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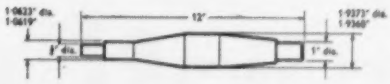
THE NEW SCRIVENER

No. 3 Centreless Grinding Machine

For continuous high production of precision ground parts the Scrivener feature of controlled cycle plunge grinding offers special economy unattainable by any other grinding method.
This example illustrates typical production.



Stub Axle.	Plunge grinding two dia. (±0.0002") and two tapers (to gauge) magazine feed.
Total time.	30 seconds. 120 pieces per hour. Stock removed 0.012" to 0.015" in two passes.

The largest machine in the Scrivener range, this new Centreless Grinding Machine was introduced at the International Machine Tool Exhibition, and received wide approbation.

Employing the Scrivener system of controlled cycle operation, it demonstrated the plunge grinding of a stub axle in a total cycle time of 30 seconds (for two operations).

Scrivener machines incorporate, in addition to features for straight-through or hand-operated plunge grinding, a system for semi- or fully-automatic controlled cycle operation. By adopting the latter method, very economical mass production rates can be achieved, the machines feed, grind and eject in sequence and can be incorporated into automatic production lines.

The complete range comprises:

Model	No. 1D	No. 2	No. 3
Grinding wheel dia. and width*	12" × 3"	20" × 6"	20" × 10"
Control wheel dia. and width ...	7" × 3"	12" × 6"	14" × 10"
Max. opening between new wheels...	2½" *2"	6" *4½"	9" *9"

*Machines with controlled cycle operation.

WICKMAN LIMITED

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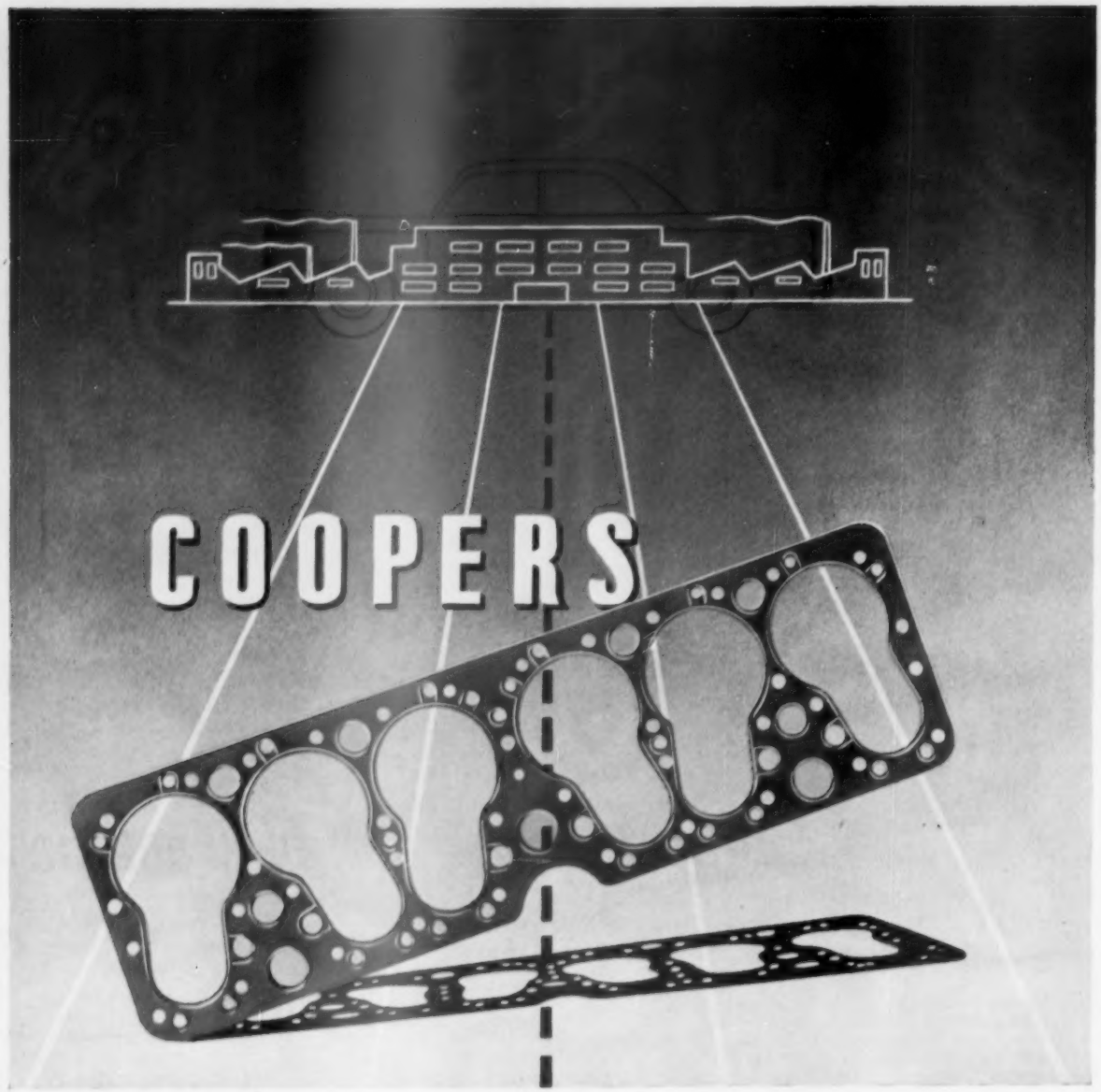

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P·B·M CASTINGS ARE USED ...

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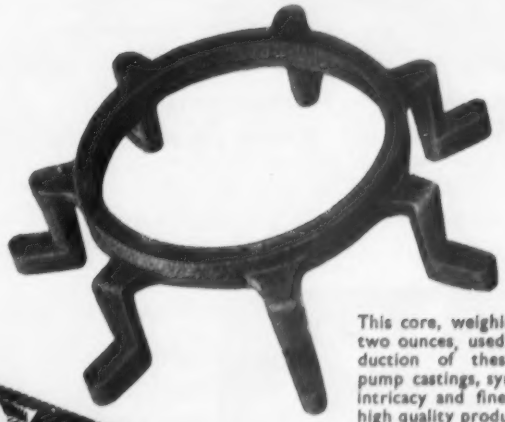
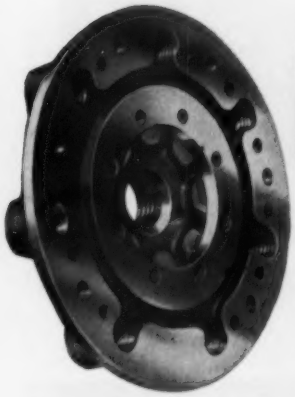


This core, weighing less than two ounces, used in the production of these hydraulic pump castings, symbolises the intricacy and fineness of this high quality production.



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MACHINED CASTINGS & ASSEMBLIES . . .

P·B·M have excellent facilities for supplying castings as fully machined and assembled components.

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MICHIGAN TOOL COMPANY

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Rate of Feed	...	**Adjustable; reduces automatically for finishing strokes.**
No. of Strokes	...	**Varies with depth of form cut.**
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Operation	... ...	**Push-buttons.**
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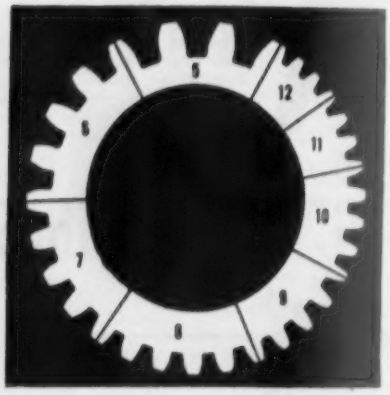
Clamping	... ...	**Hydraulic or mechanical.**
Cutter Construction		**Formed tools.**
Tool Sharpening	...	**Surface grinder.**



Illustrated (reduced size) on left are the pitches of gear teeth that can be cut on the SHEAR-SPEED. The versatility of the machine is indicated by the fact that the entire form illustrated could be cut in one operation, all the teeth of different pitches being formed simultaneously.

GEAR STEEL CHIPS AT....

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The SHEAR-SPEED cuts all teeth of a gear or external forms simultaneously, in a small fraction of the time required by conventional cutting processes.

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LO-4

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LO-2

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IN THE FOLLOWING PAGES WILL BE FOUND ITEMS OF PARTICULAR INTEREST FOR

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the Lockheed Brakemaster has become established as the finest development of the conventional expanding brake. The brake has outstanding stability and performance under the most severe conditions, and with just the right degree of assistance from its servo unit.

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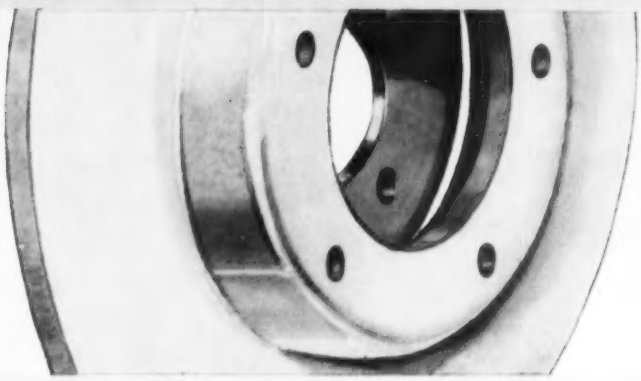
BRAKEMASTER: Trade Mark

LOCKHEED

LOCKHEED: Regd. Trade Mark

DISC BRAKE



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embodying the experience of many years of disc-brake development. A fine, powerful brake giving a consistent performance, with the generous lining thickness essential to this type of brake. Simple, compact and easy to build into both the rear and the front end.

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LOCKHEED: *Regd. Trade Mark*

LOCKHEED

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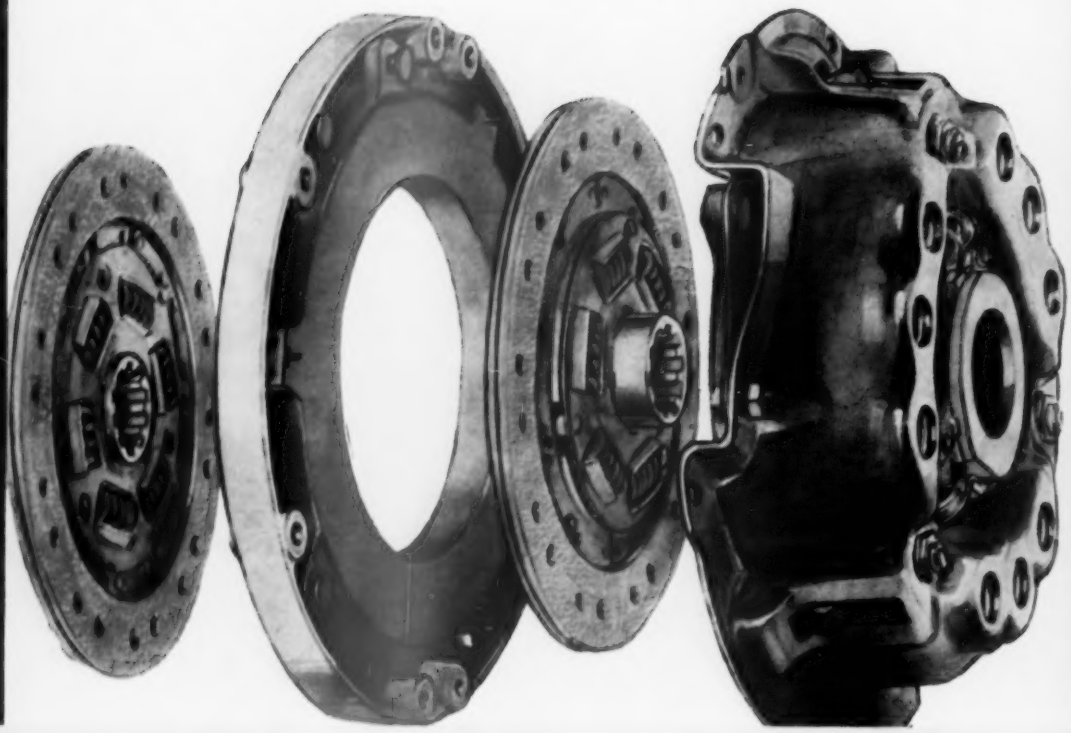


The present trend towards high performance lower-built cars—and the corresponding reduction in fly-wheel diameter—is catered for by the Borg & Beck 8 in. strap-drive clutch which has a high torque capacity in relation to diameter—250 lb./ft., and a safe speed of 9,000 r.p.m.

It is also available as a single plate strap-drive version—torque capacity 125 lb./ft.

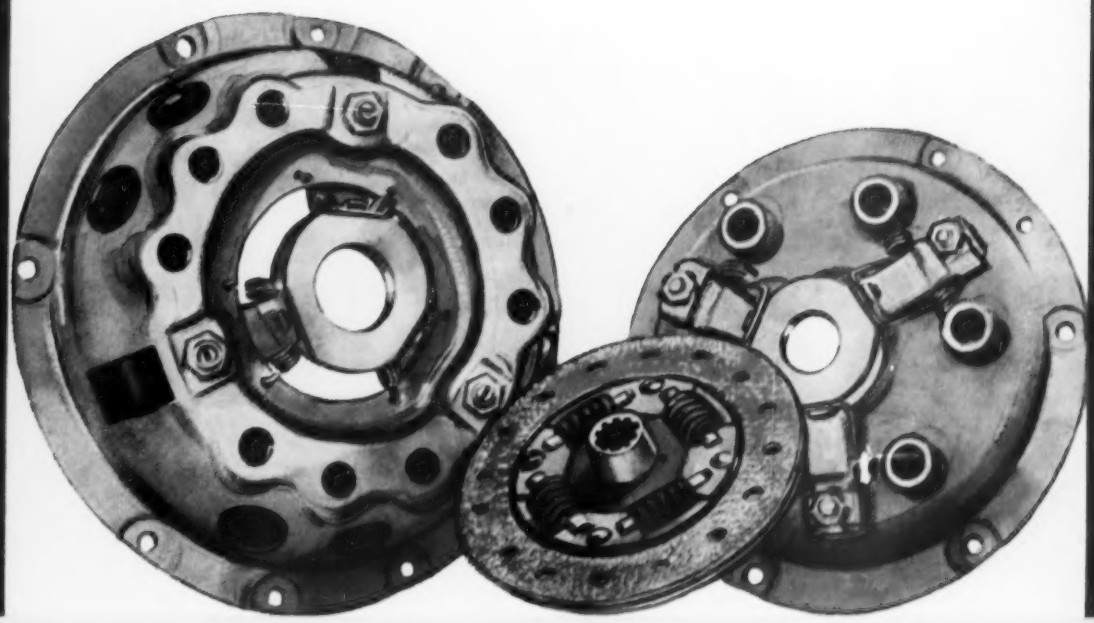
The successfully proved Borg & Beck tangential strap-drive method of carrying the pressure plate on the clutch cover has the advantages of giving a silent drive, low frictional losses, and a consistent sense of balance in the clutch as a whole.

Either a carbon or a ball-bearing release mechanism may be used.

Fully Patented

BORG & BECK

Regd. Trade Mark: Borg & Beck



The famous 'A' type range of Borg & Beck single dry-plate clutches embraces the 6¼ in., 7¼ in., 8 in., 9 in. and 10 in., having torque capacities of 38, 55, 80, 130 and 150 lb./ft. respectively.
(The range extends beyond this into the 11 in., 12 in. and 13 in. sizes used on commercial vehicles, and the 'R'-type single and twin plate 14 in., 16 in. & 18 in. clutches, the largest having a torque capacity of 1600 lb./ft.)
Borg & Beck clutches are fitted to over 80% of Britain's vehicles and embody many features which have become famous for their influence on smooth performance and durability.

BORG & BECK COMPANY LIMITED,
LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

Regd. Trade Mark: Borg & Beck

BORG & BECK

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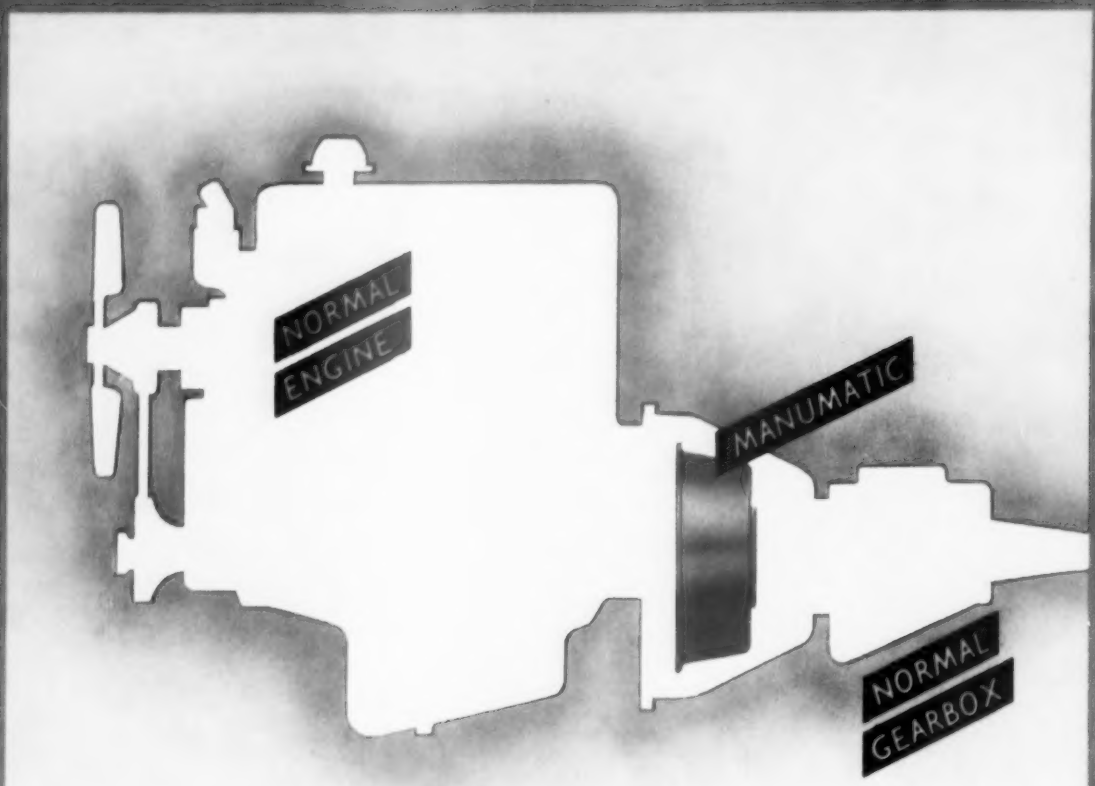

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Selection MANUAL: *gear change* AUTOMATIC

The only transmission system that combines these advantages:—

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3. ***Utilizes normal synchromesh gearbox.***
4. ***Embodies Borg & Beck centrifugal clutch.***

MANUMATIC—well proved under prolonged usage—is the simplest for a manufacturer to incorporate, and the simplest, safest and most pleasant to drive. The Manumatic system uses the normal synchromesh gearbox and is ideal for the popular type of British car.

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earned their fine reputation by their unequalled efficiency in removing the microscopic particles from the engine oil.

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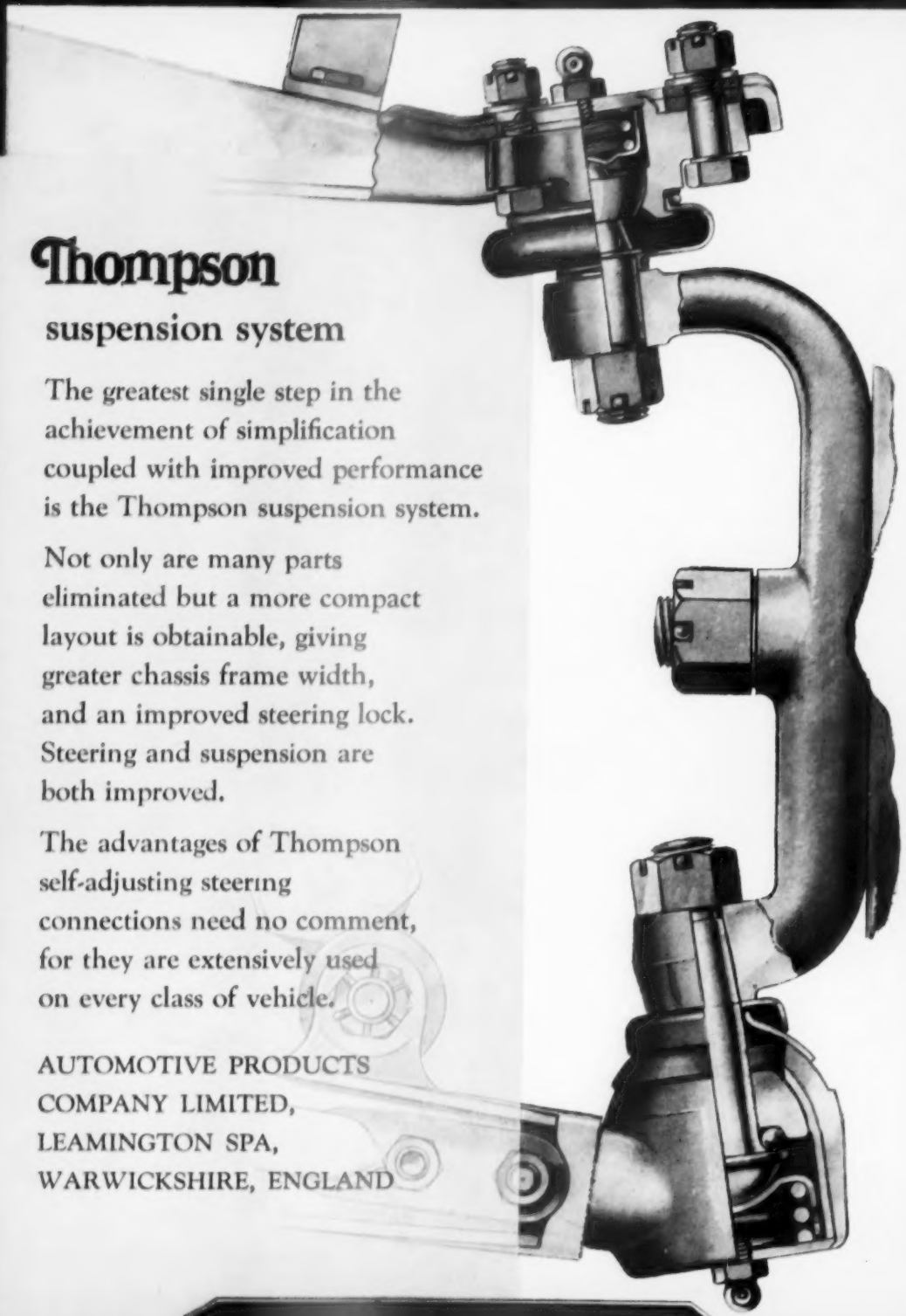

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Not only are many parts eliminated but a more compact layout is obtainable, giving greater chassis frame width, and an improved steering lock. Steering and suspension are both improved.

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The Motor Show's Top News

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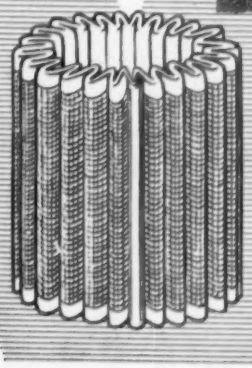
For the sake of the first few minutes . . . and of the years of service that follow, specify Vokes filters.



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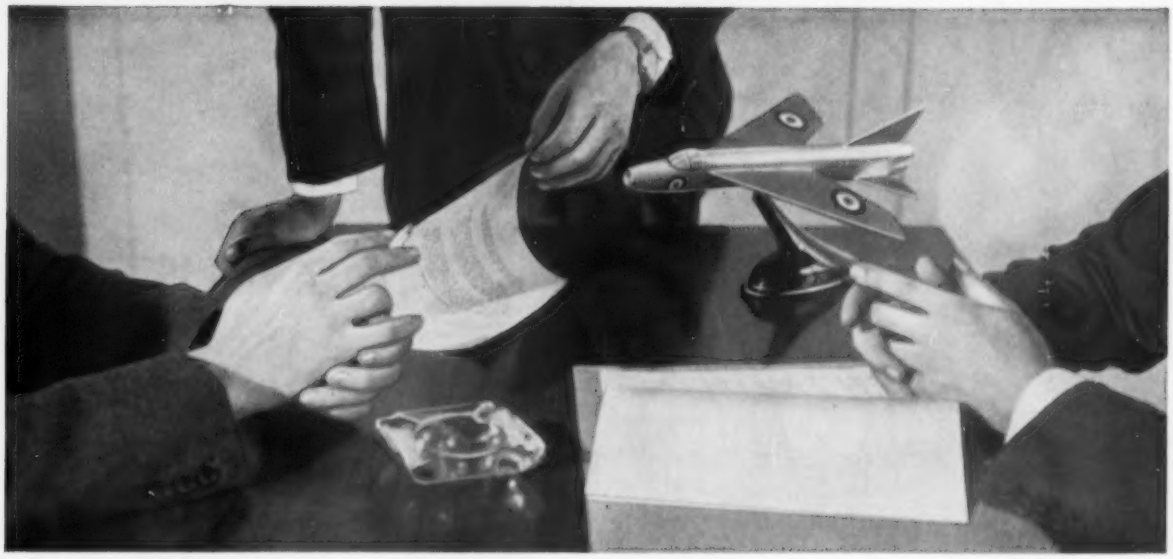

'If we can't find more men from inside the aircraft industry— *then we must look elsewhere . . .*'

(Report of a meeting between the Chief Engineer, his Technical Assistant and his Personnel Officer)

C.E. As I see it, the problem is how to attract people by conveying some idea of our future plans without giving away secret information.

P.O. Haven't we first got to convince them that ENGLISH ELECTRIC intend to push ahead with the next generation of aircraft?

C.E. Surely they don't think that after all the work we've done on supersonic flight in general, and the P.1. in particular, we are just going to sit back and go out of business!

P.O. If we can't talk about the aircraft we are planning to build can we say something about the equipment we've got or are going to get?

T.A. For example, is it permissible to point out that our new tunnel has been designed for speeds up to Mach 4, and let people draw their own conclusions?

C.E. There's no reason why we shouldn't. And we can mention the new plant for testing structures and equipment at high temperatures non-stop for hours at a time. In fact we ought to talk about all our test installations.

P.O. Provided we don't thereby give people the impression that we are a great octopus in which the individual is swallowed up.

T.A. I've never yet heard a research man complain about the amount of equipment or the size of the resources at his disposal!

C.E. True, but it's worth pointing out that this Division is a self-contained affair of very reasonable size, with a colossal parent organisation behind it.

P.O. Which of course brings in all the other advantages like security of employment, opportunity for advancement, decent working conditions and help with the accommodation problem.

C.E. And remember we want all grades and sorts —mechanical and hydraulic engineers, heat-transfer experts, metallurgists, physicists, aerodynamics folk—the lot. We must look outside the aviation industry as well as inside. See if you can get something on paper for me.

'ENGLISH ELECTRIC' aircraft

Partners in Progress with Marconi's and Napier in THE ENGLISH ELECTRIC GROUP

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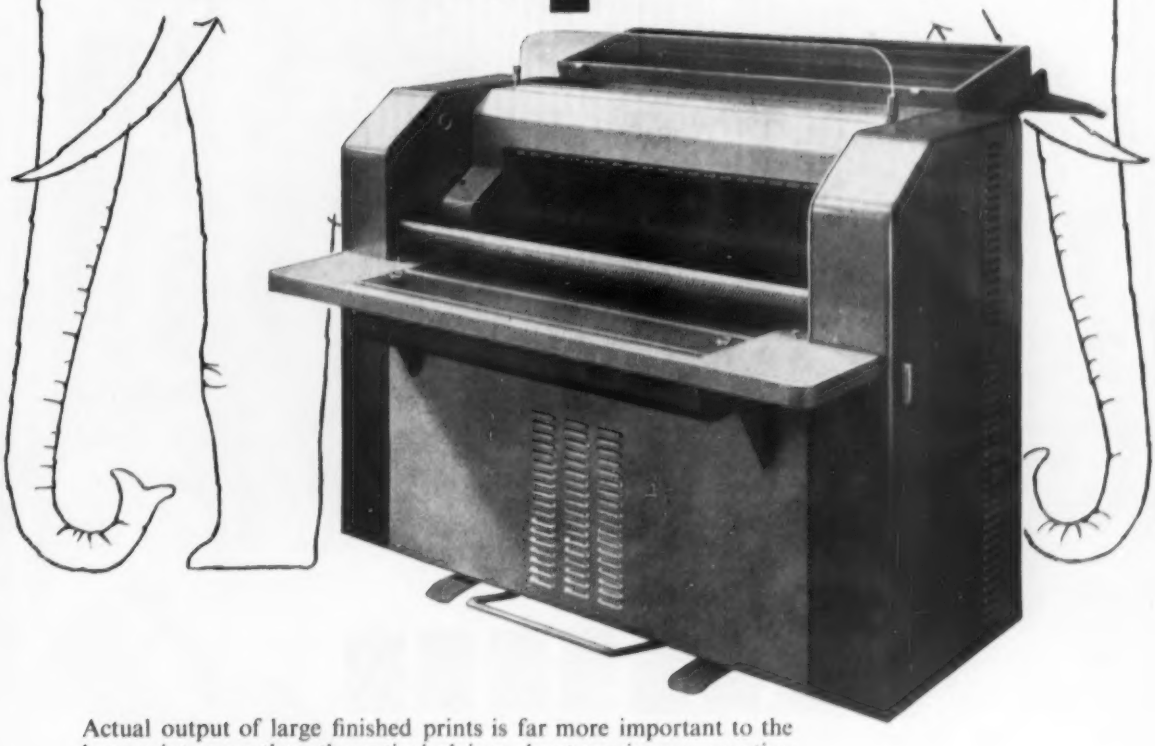
How many ? double elephants



Actual output of large finished prints is far more important to the busy print room than theoretical claims about maximum operating speeds. Viewed from this angle, the performance of the AZOFLEX Model 46/35 Combine Printing and Developing Machine is altogether outstanding, for it is capable of turning out prints at a rate well in excess of 1,000 Double Elephants a day, year in and year out. Exposure, development and print delivery are all synchronised for an even flow of finished work and optimum quality is always ensured because Azoflex is the only photo-printing process to apply the correct quantity of developer, irrespective of running speed. In addition, the process is free from unpleasant fumes and requires no complicated ventilating system.

For use with the Model 46/35 and other machines in the Azoflex range, there is a wide choice of high-quality Azoflex materials designed to meet every photo-printing requirement.

AZOFLEX MODEL 46/35
Synchronised printing and developing machine. Capacity: cut sheets and rolls up to 42 in. wide. Printing speed: from 2 ft. to 30 ft. per minute. Dimensions: height 58 in., depth 60 in., width 72 in. Weight 1,100 lb. Price £1,555 including delivery and installation in the U.K.

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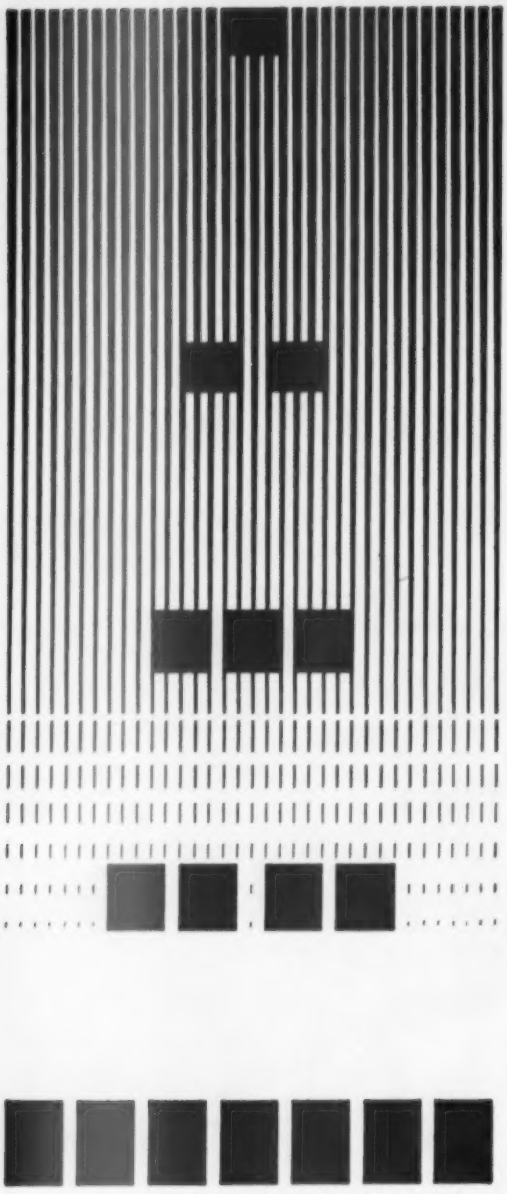
ILFORD *Azoflex*

PHOTO-PRINTING MACHINES AND MATERIALS



Production grows with good lighting

IN THE MAJORITY of works today the lighting is inadequate or unsuitable, or both. Production is slower than it could be, mistakes are made, health and tempers suffer.

Money spent on a planned lighting system is the best of investments. The foreman can see how things are going; the craftsman at his machine can work more accurately and easily because he can see what he's doing.

Good lighting is only one of the ways in which electricity is playing a vital part in the drive for higher productivity.

Electricity for Productivity

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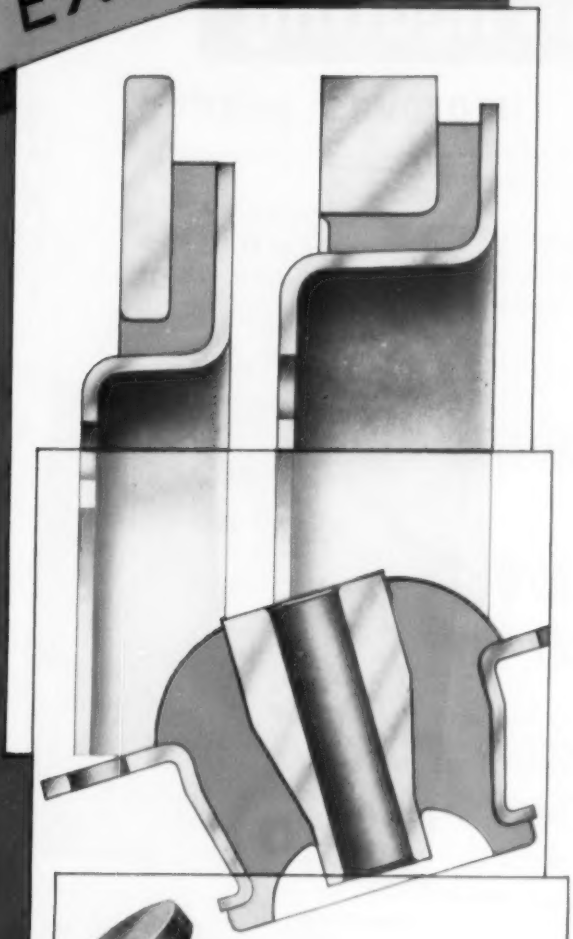

Cumulative benefits from Metalastik specialization

Every Metalastik component—whatever its function—benefits from the experience we have gained in solving previous problems, and in its turn paves the way for further advances.

Metalastik leadership is the outcome of this deep-rooted specialization and has established rubber-bonded engineering as a necessity to the motor industry.

These components are typical of Metalastik leadership :

TORSIONAL VIBRATION DAMPERS

No component in the world is so simple and yet so efficient in performing an important duty as the Metalastik torsional vibration damper. By analysing the requirements, Metalastik can design and manufacture dampers to suit any engine.

"METACONE" MOUNTINGS

For engines, bodies, cabs and many other diverse applications Metacone mountings are another example of Metalastik success. With the rubber-to-metal weld to both inner and outer sleeves, creep is prevented and vibration-absorbing properties remain constant in service. Available in many designs for a wide range of loads and deflections.

HIGH-DUTY AND ULTRA-DUTY SHACKLE PINS

The excellent performance of these shackle pins which now have millions of trouble-free miles behind them is endorsed by their continued and growing use on goods and passenger vehicles.

METALASTIK

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BIRLEC

cuts costs

induction heating

improves product

☆ The BIRLEC induction surface hardening machine illustrated here is a standard unit capable of handling a range of varied jobs. It is widely used by the automobile industry for the selective treatment of rocker shafts, selector rods and similar components.

VERSATILE SHAFT HARDENING MACHINE

Specified zones can be hardened under automatic control and with negligible distortion or scaling of the work. Quickly interchangeable, standardised control cams and inductors permit several different jobs to be run economically in successive batches.

Illustrated are typical induction hardened automobile rocker shafts and selector rod.

Maximum length of work	24"
Maximum diameter of work ...	1½"
Hardening zone selection	Cam
Maximum hardening speed	1½"/sec.
Maximum return stroke speed ...	6"/sec.
Typical production (12" stroke) ...	300/hr.
Typical operating cost ...	0.17d/piece.

Please send me Publication No. 78 on Shaft Hardening or further process details about

.....

Name .....

Position .....

Company .....

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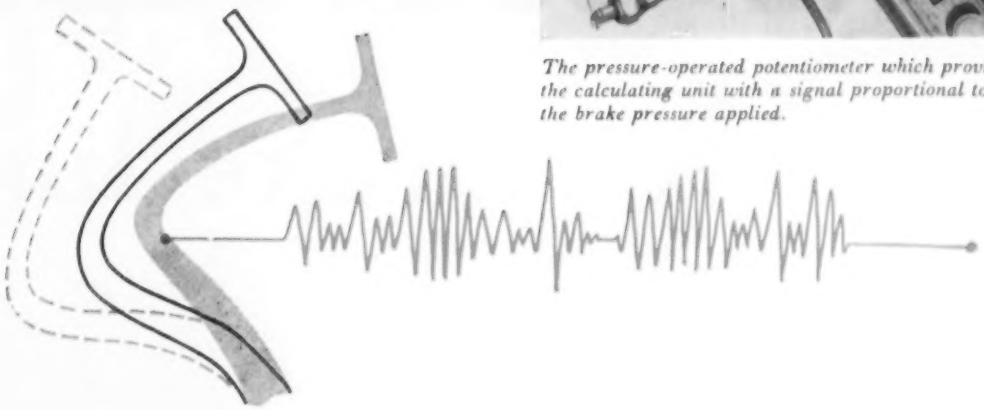
BACKGROUND TO BRAKING No. 1

The pressure-operated potentiometer which provides the calculating unit with a signal proportional to the brake pressure applied.



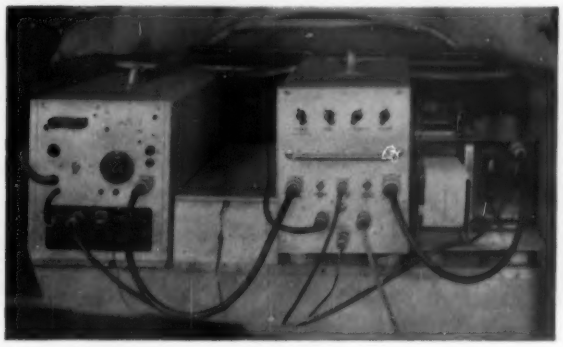
Electronics aid the search for Safety

In devising schedules for tests, Ferodo research workers must know what is expected of a brake lining under practical conditions. Without this information it is possible to overrate the lining—make it break down in a manner which would not occur in service.

The performance of a brake lining depends mainly upon the rate at which work is done at the brake and upon the temperature of the drum surface. To record this information Ferodo technicians have installed ingenious apparatus on a vehicle of the Ferodo Test Fleet.

A tachometer generator, driven by the vehicle's propellor shaft provides a signal which is used to record the speed and the deceleration of the vehicle on a high-speed multi-pen recorder. A voltage proportional to the product of the hydraulic pressure in the brake system and the speed signal is applied to a third pen. Subject to certain necessary precautions, in particular that the brake factor should be frequently checked experimentally, this product is a measurement of the rate of working at the brakes. A thermister is soldered to the surface of the drum to record temperature on a fourth pen.

The results enable Ferodo to devise testing schedules that are accurate and reliable and so to produce brake linings with a high resistance to fade and wear.



Some of the apparatus in the back of a test car. The power pack is on the left and the chart can be seen emerging from the four pen recorder.

FERODO

ANTI-FADE Brake Linings

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with CAPASC

moulded brake linin

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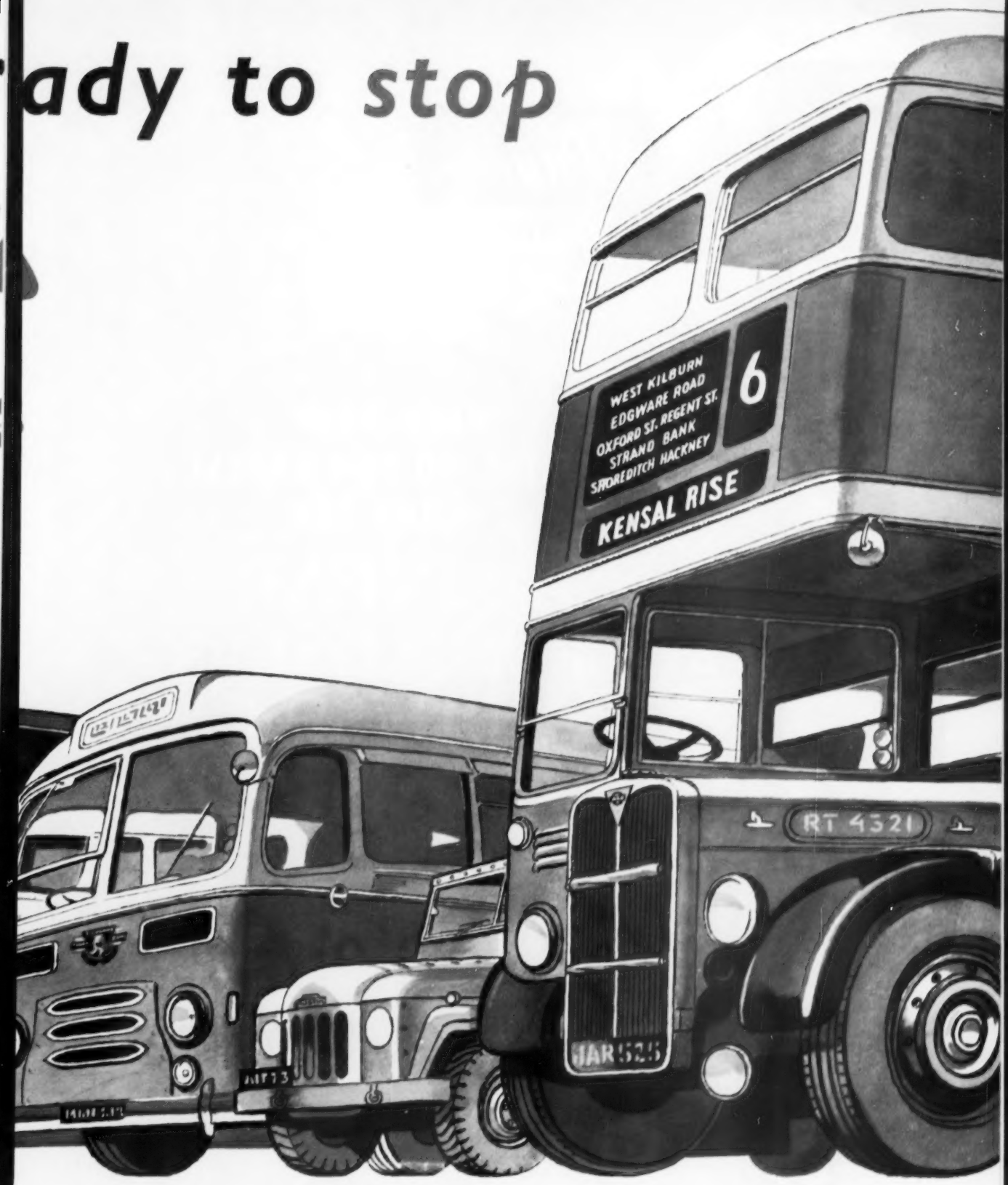

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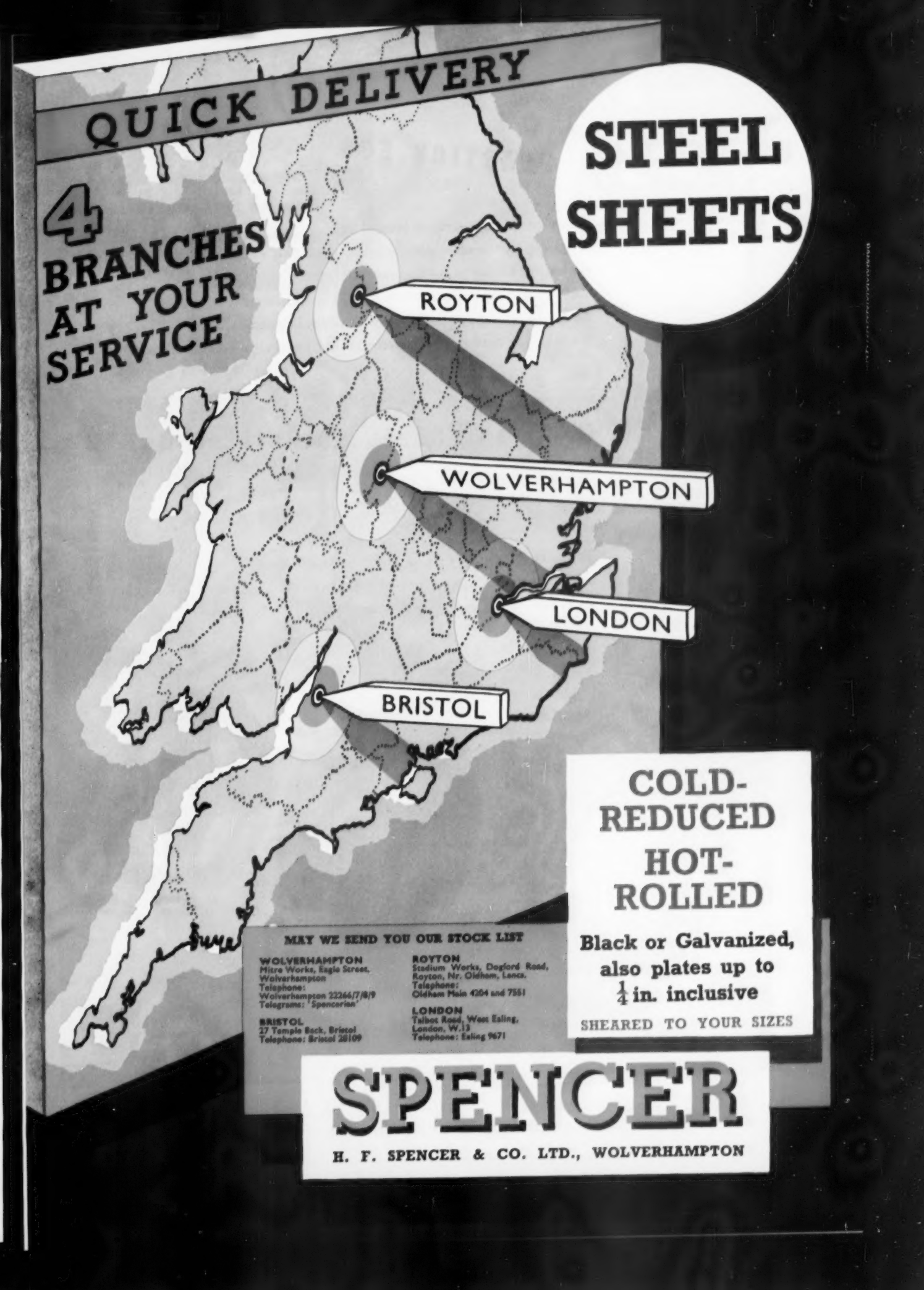
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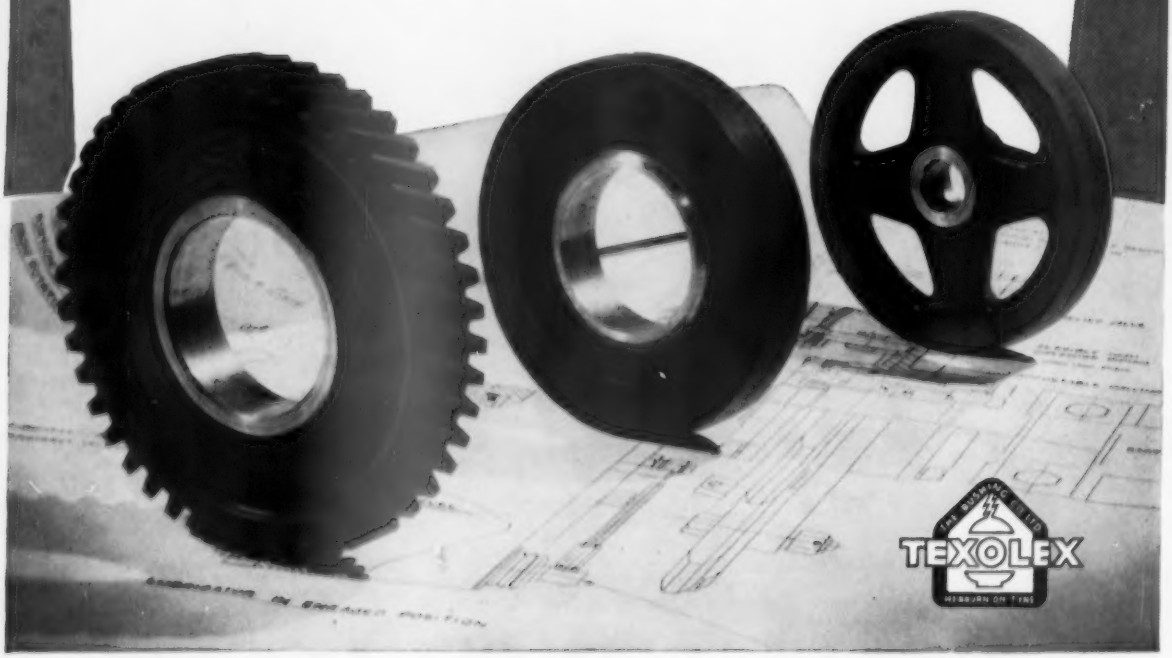
TEXOLEX IN THE AUTOMOBILE INDUSTRY

The Bushing Company Limited, would be pleased to give further technical information about Texolex laminated fabric gears and their suitability and operation in camshaft, jackshaft and oil pump drives.



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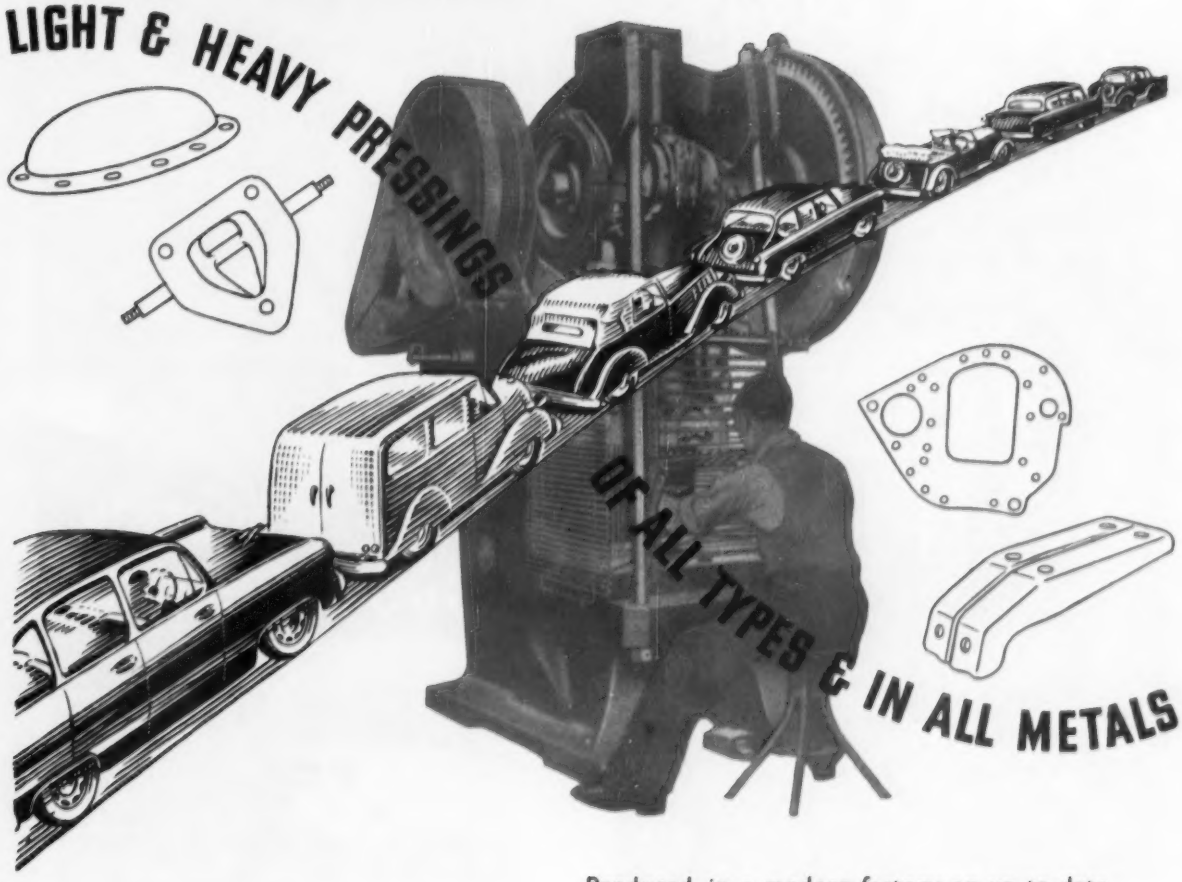

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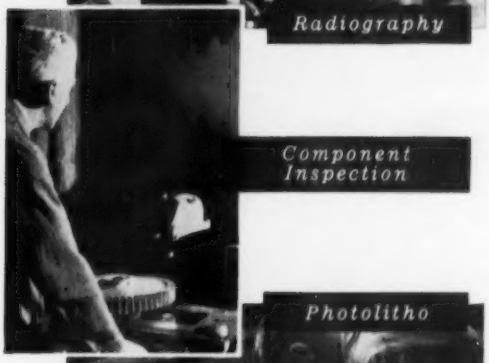

Component Inspection

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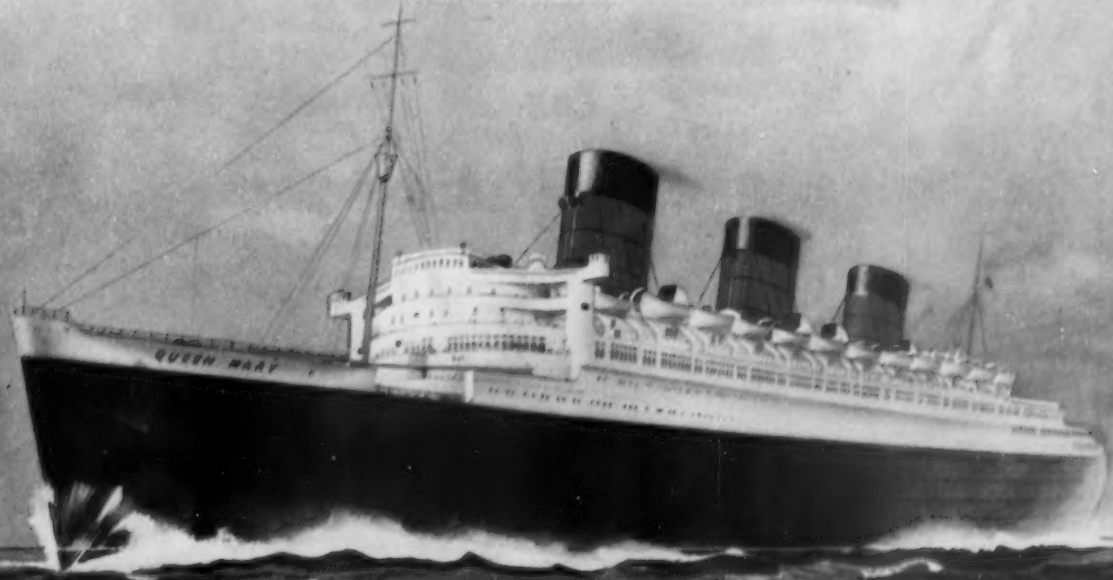

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The reason is that Molybdenum makes it easier to ensure the following combination of advantages:

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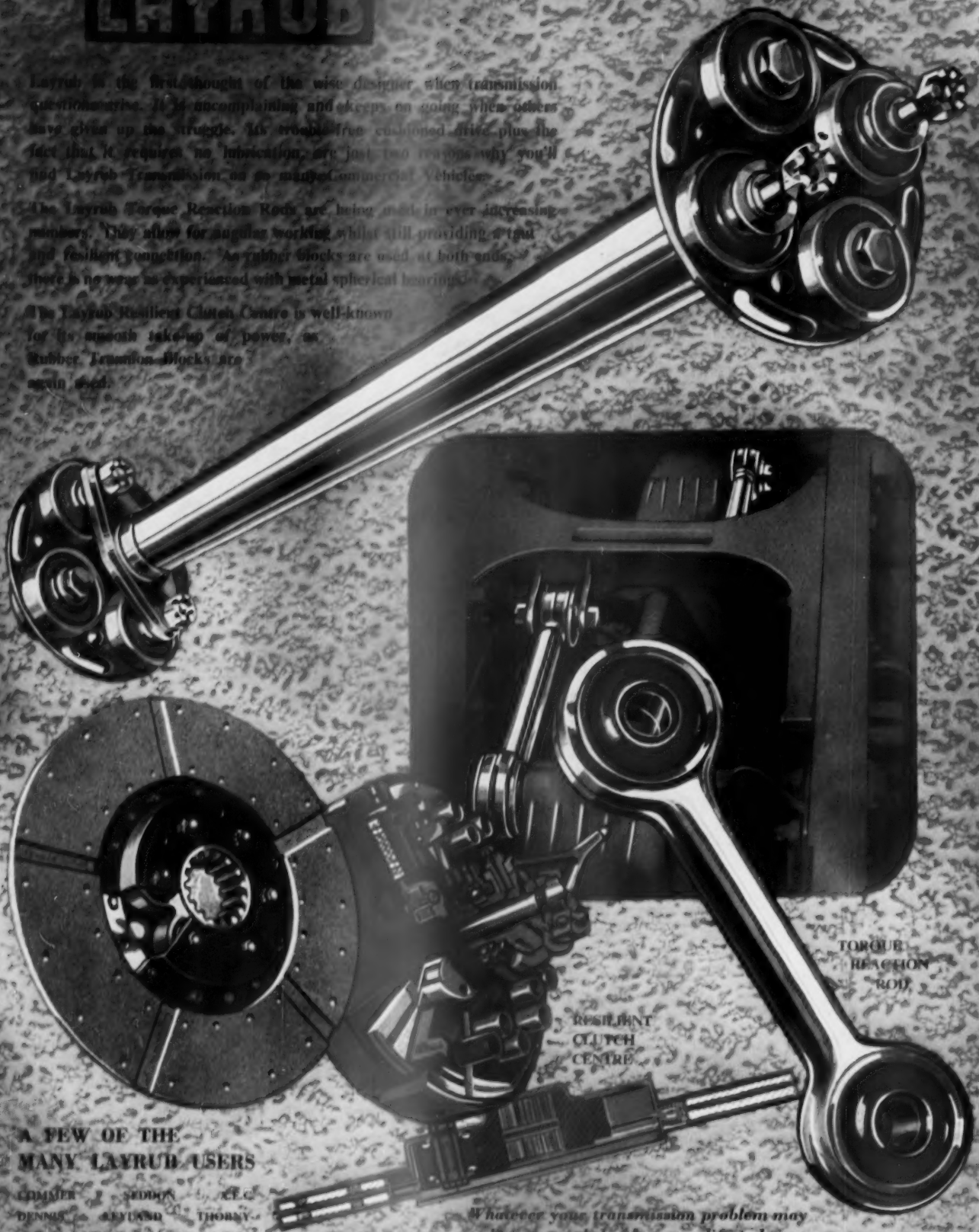

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MAXIRAPID

AUTOMATIC MULTI-TOOL LATHE

designed specifically for fully automatic high output production

Top illustration shows loading arm ready for automatic exchange of components between centres.

Lower illustration shows slides in cutting position. Unmachined components can be seen at the front of the loading unit. Machined parts are ejected at the rear of unit.

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The illustrations above show the "Maxirapid" tooled up for turning forged steel final-drive pinions. This example demonstrates the possibilities of the machine for really high speed heavy duty production.

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The illustrations show the machine with splash guards removed.

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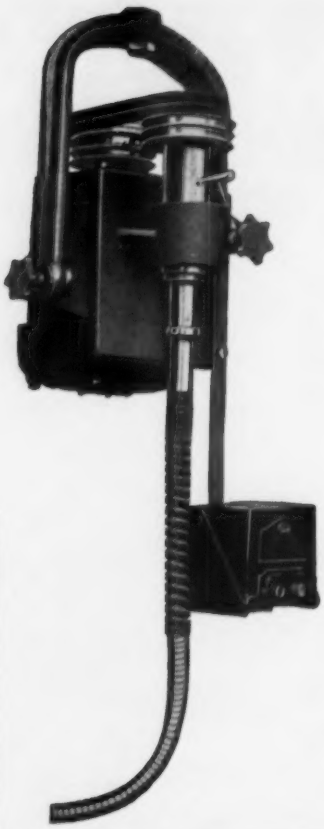

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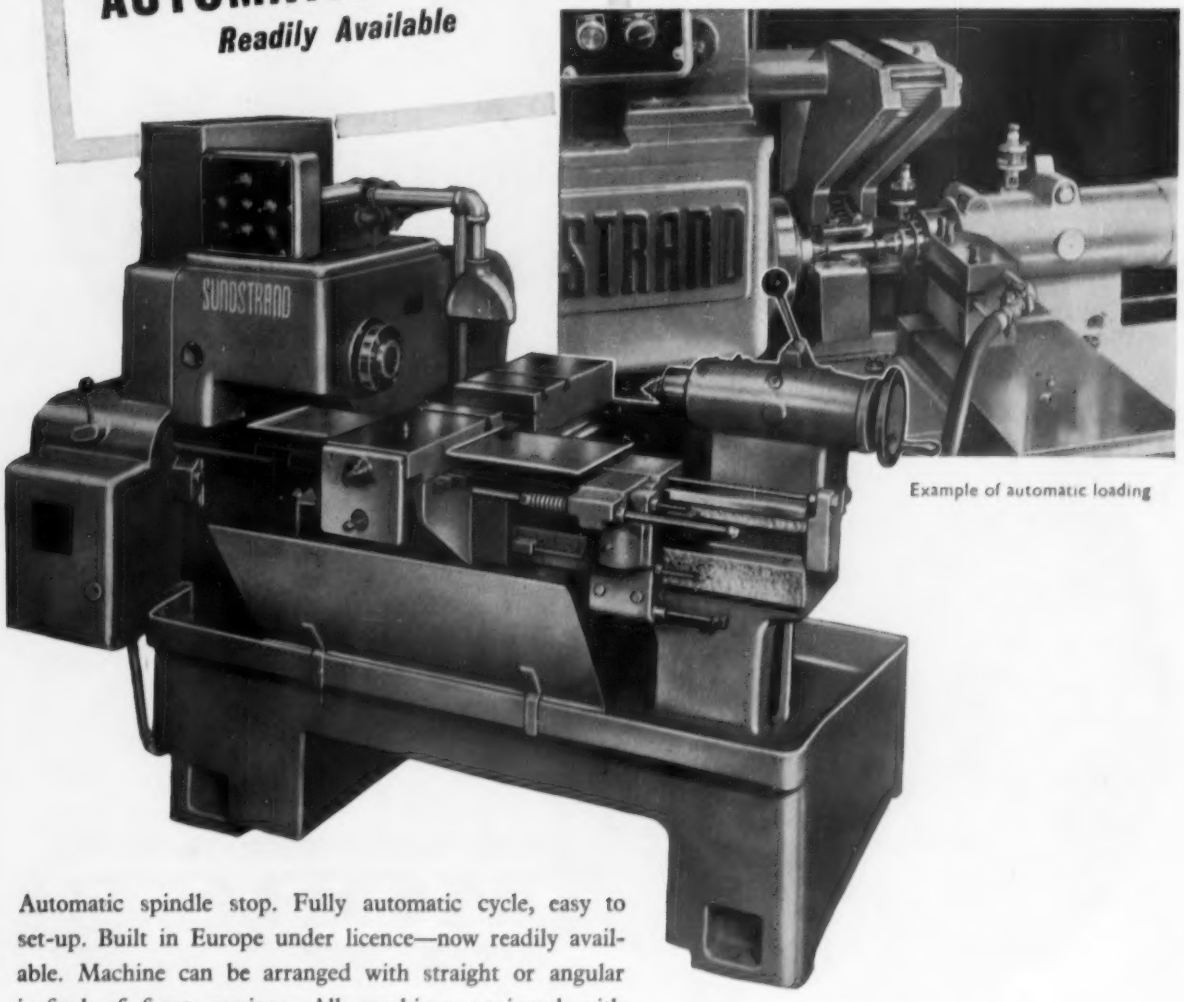

Example of automatic loading

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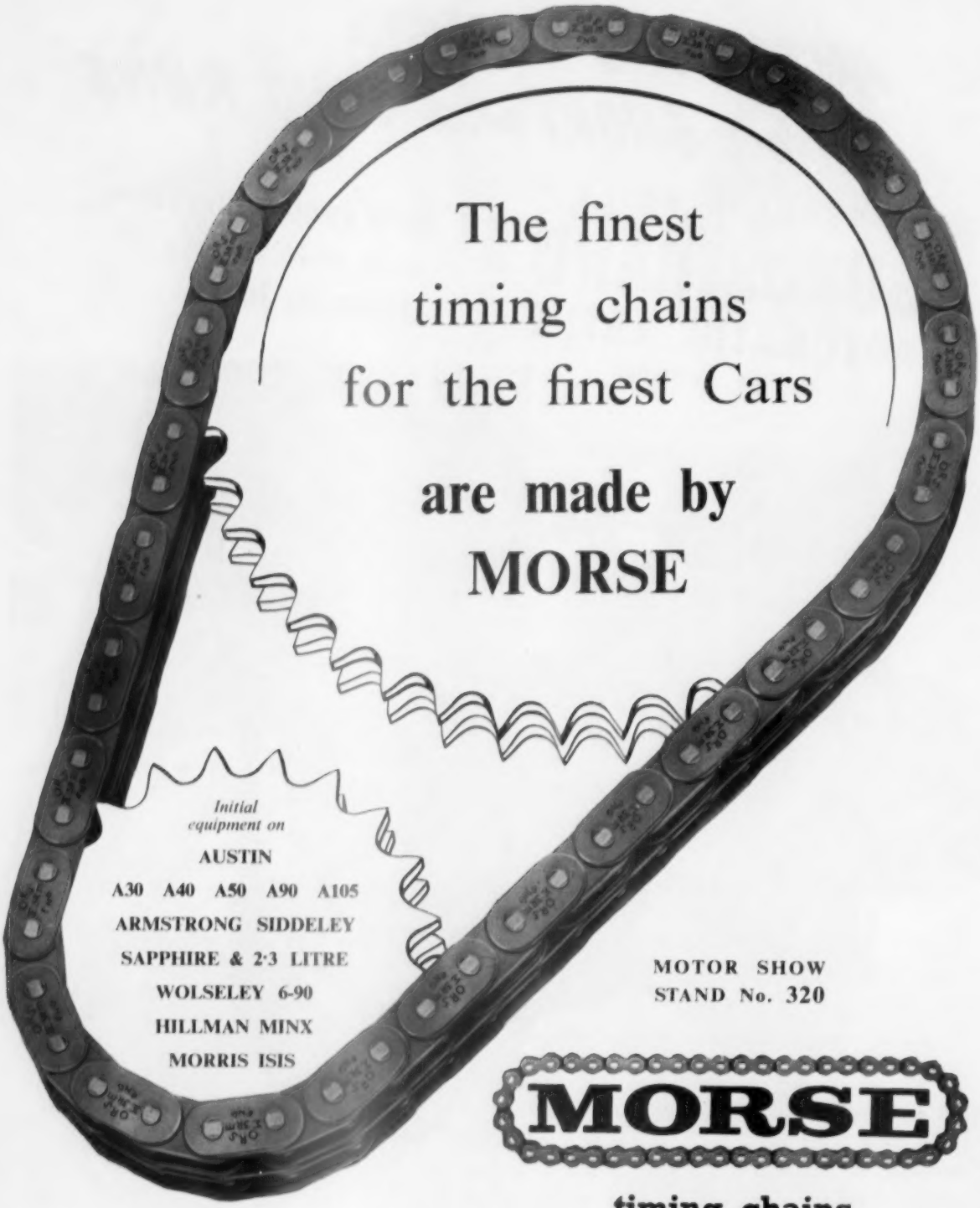

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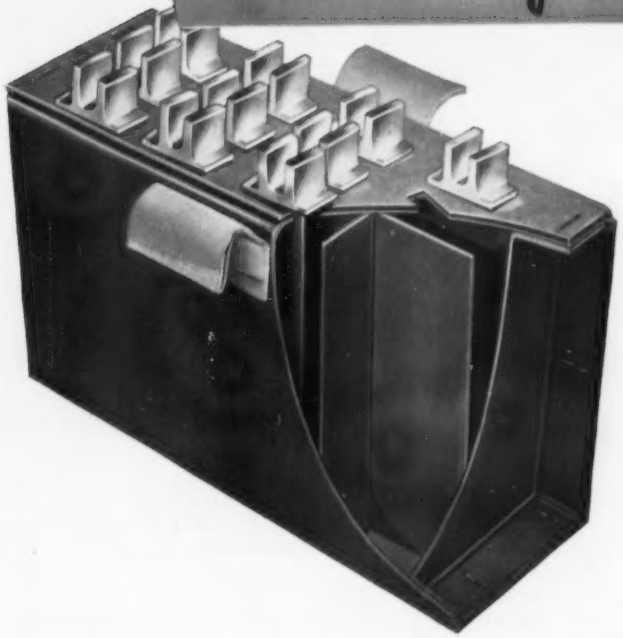
B-W

Sheathing the compressor blade for ROLLS-ROYCE



When Rolls-Royce needed a "cradle" to protect their compresser blades during the stages of manufacture, the answer was a strong, double-walled cardboard box designed by Robinsons. These economical boxes have special individual compartments which keep the blades secure from abrasion during handling.

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This new approach to the handling of spares is typical of Robinsons' application of better packaging and printing techniques in almost every Industry. If you feel that your product is handicapped by outmoded dress, we invite you to take advantage of Robinsons' skill, ideas and resources.

R

Pouches and wrappers
Bags and envelopes
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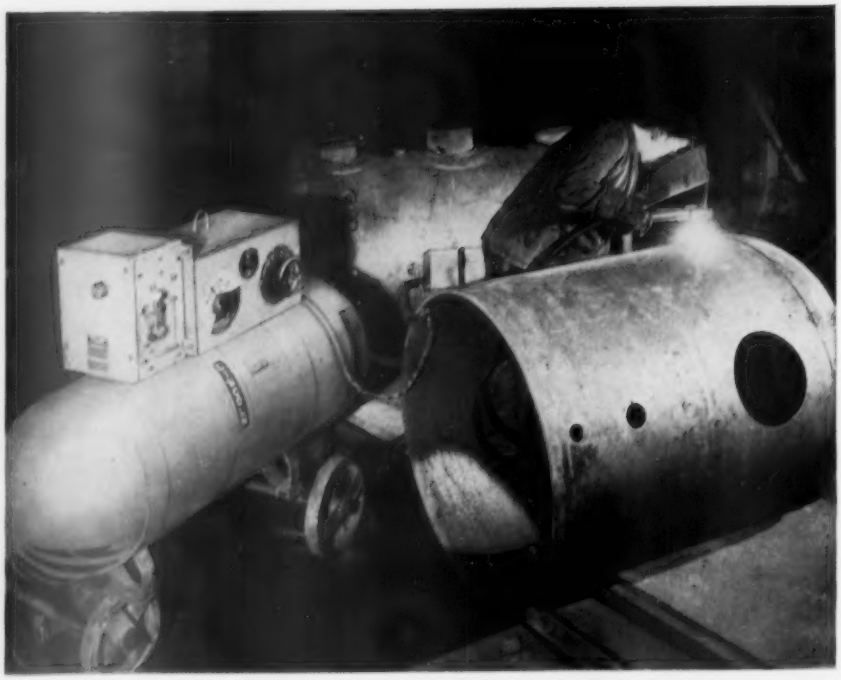

The Empyrium Welding and Manufacturing Co., Ltd of Birmingham cut and gouge with the Arcair torch, here seen being used on Air Receivers for the Automobile Industry. Cutting back and gouging welds is seldom necessary because, as they say, 'they have an excellent bunch of welding operators who do a 100% job with Lincoln Electrodes and Lincolnweld Automatic Welding equipment'. Nevertheless, they recommend this 'excellent tool' when the occasion arises.

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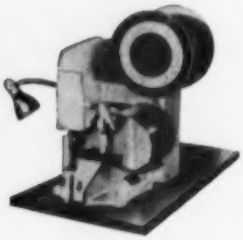

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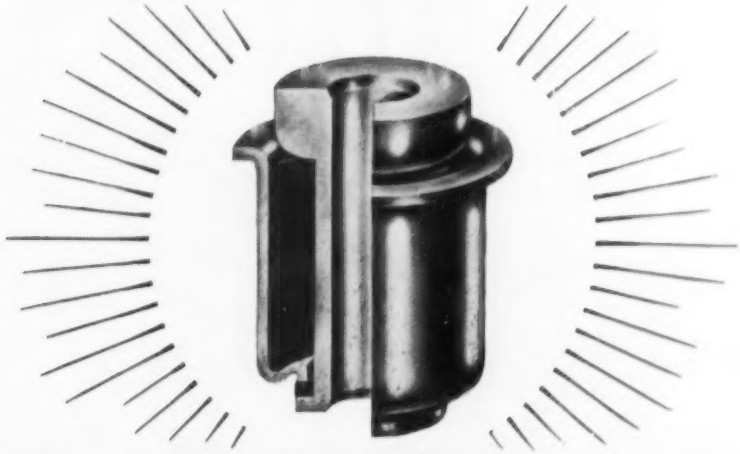

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● Bus bodies last longer when protected from vibration by Silentbloc flexible mountings. Of robust construction these units need no attention. Vibration is absorbed by a scientifically designed rubber insert which is protected from damage by a steel casing. Silentbloc mountings in use on double-decker buses were as good as new after 250,000 miles. Our Technical Department will be glad to submit mounting schemes.

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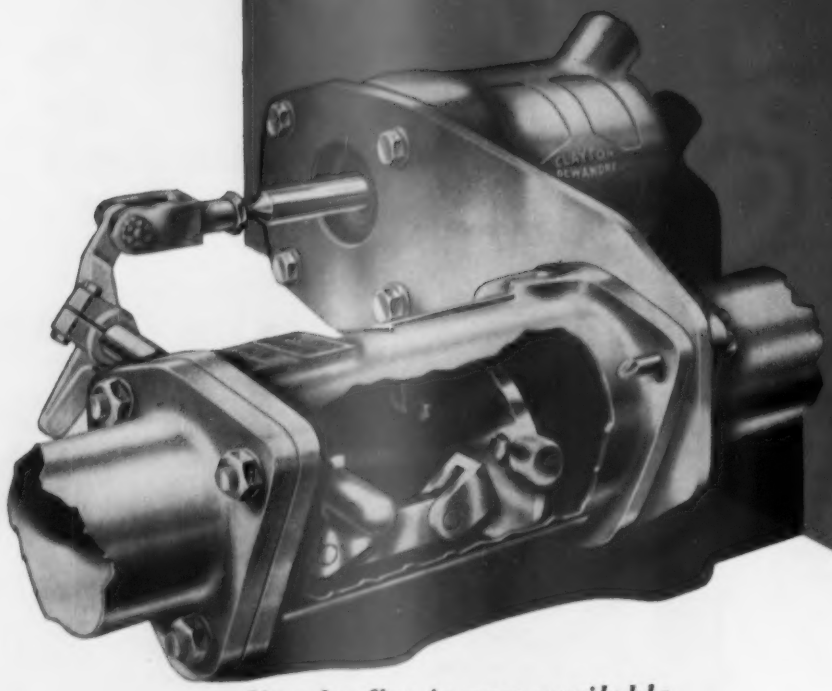

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The control of quality and cost of any engineering product starts in the standards room.

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Specification E393.22, giving complete details, is available on request.

STANDARD GEARBOX RANGE

	Type	Max. Engine Torque (lb/ft)	No. of Speeds
	430C	90	4
	437	133	4
*	542	205	5
*	45	250	5
*	045	250	5
†	S450	300	4
*	557CM	350	5

2-speed epicyclic unit—either overdrive or underdrive, suitable for 350 lb. ft. maximum engine torque.

* *These units are available with either an overdrive or direct top gear.*

† *Synchromesh for all speeds.*

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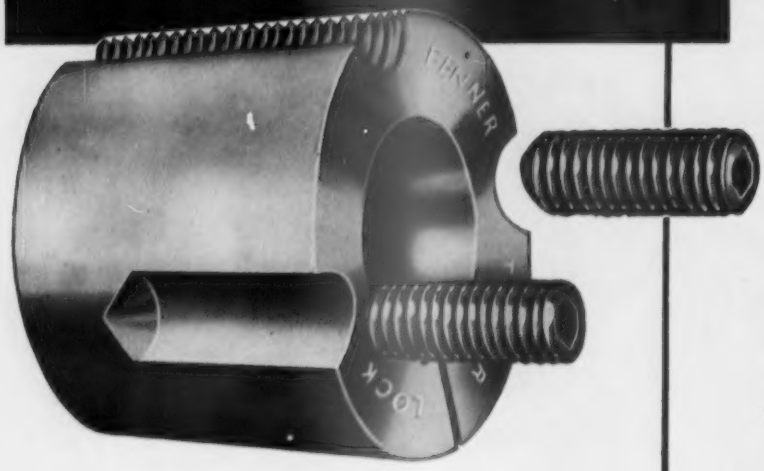
DAVID BROWN
CORPORATION (SALES) LIMITED
AUTOMOBILE GEARBOX DIVISION
PARK WORKS HUDDERSFIELD

500,000 without a failure!



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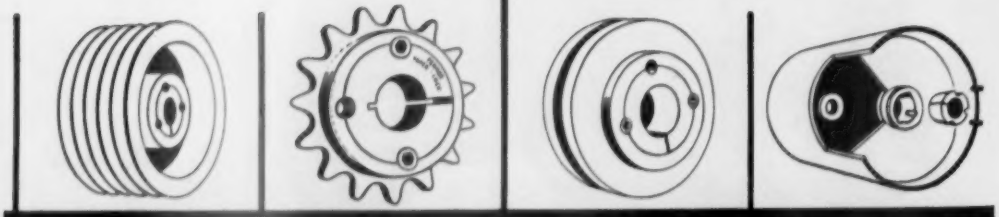

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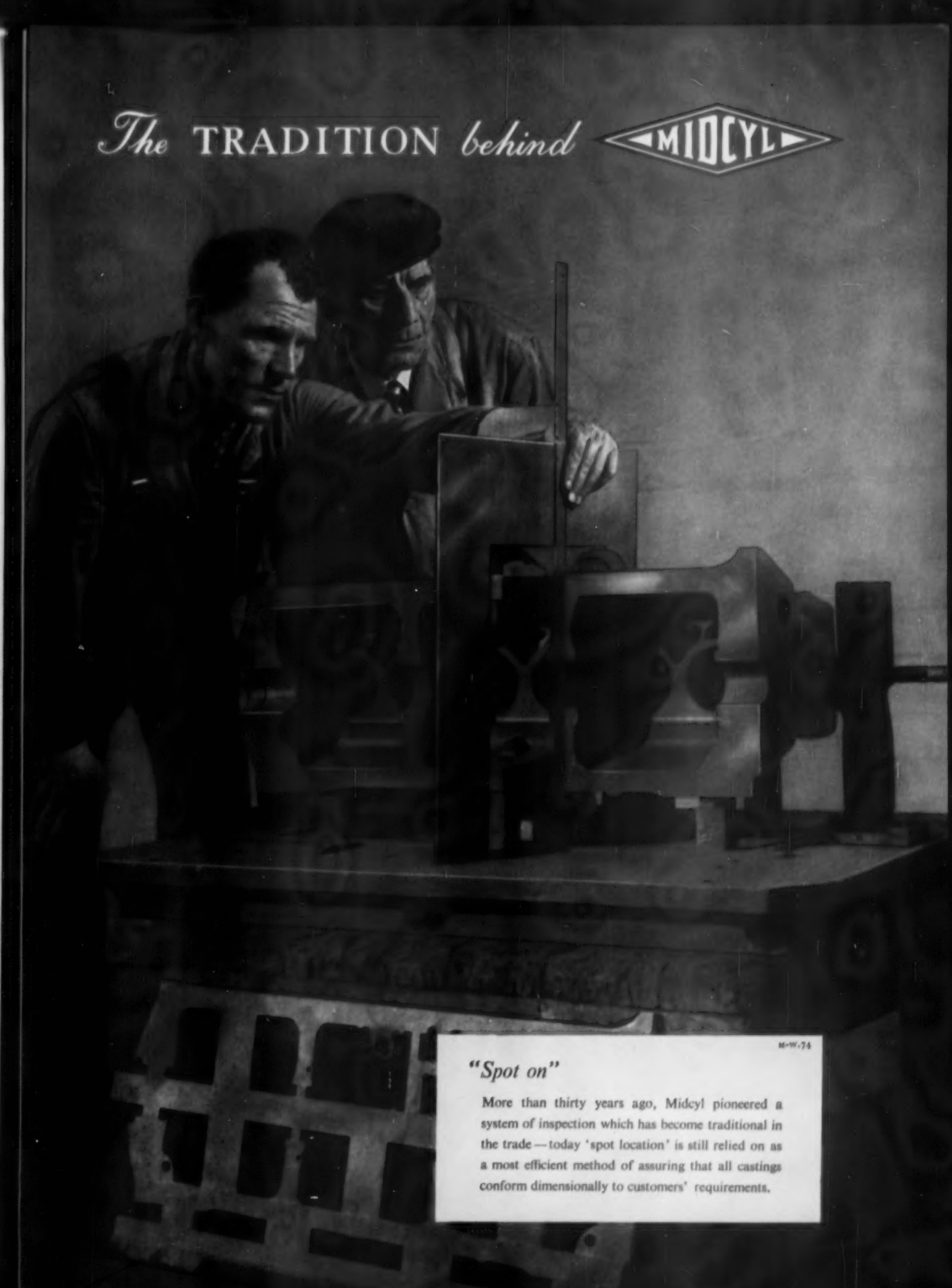
The TRADITION *behind* MIDCYL



M-W.74

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More than thirty years ago, Midcyl pioneered a system of inspection which has become traditional in the trade—today 'spot location' is still relied on as a most efficient method of assuring that all castings conform dimensionally to customers' requirements.

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FBC SPECIALS

. . . the key to a fuller life

Performance in many branches of engineering is getting very intensified, and what once passed as normal bearing practice is now as inadequate as a wooden wheel on a 100 m.p.h. car.

Some machines start out by packing enormous power into a small and often hot space, and there — as in some of the finest jet engines in the world — you will find FBC 'Specials', often running at fantastic speeds.

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Here again FBC 'Specials' are saving the situation — for it is amazing what exacting duties can be carried out by the right kind of bearing, when one is really 'up against it' and enlists the aid of the right people.

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Enquiries to: Laycock Engineering Limited, Overdrive and Transmission Sales Division, 16/17 Hertford Street, Coventry.
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ALL PURPOSE
Electric Ovens

FOR
DRYING, CURING, BAKING, SOFTENING, TEMPERING, BONDING, PRE-HEATING, AGEING, DE-EMBRITTLING, Etc.

FUNDITOR electric ovens have won a reputation throughout industry for their ability to meet varied requirements economically and efficiently. The range of models covers types from small bench equipments up to large batch ovens.

FUNDITOR ovens are supplied with either forced air circulation, or as natural convection ovens with temperature ranges up to 350°C. and 450°C. respectively.

Rapid and uniform heat distribution combined with precise automatic temperature control, are basic features of FUNDITOR design. Double-skin construction with fibre-glass lagging ensure minimum heat loss by conduction through steel casting.

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Although our range of standard ovens is very wide, we are always pleased to supply 'specials' to meet requirements.

Hornos eléctricos FUNDITOR para todos los fines.
Para secar, curar, hornear, ablandar, revenir, ligar, precalentar, envejecer, quitar la fragilidad, etc.
Los hornos eléctricos FUNDITOR han ganado una gran reputación en todas las industrias por su capacidad para hacer frente a necesidades variadas de una manera económica y eficaz. La serie de modelos cubre tipos desde los equipo pequeños de banco a los grandes hornos de cargas.

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The General Electric Co. Ltd,
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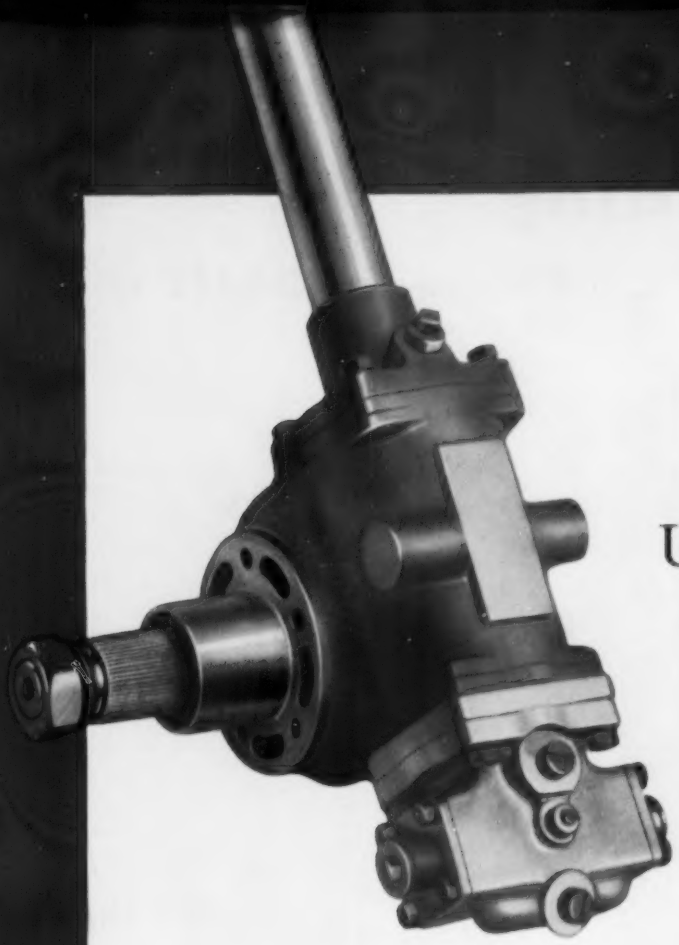

THE UNIVERSAL UNIT

In a previous advertisement we showed the Marles patent 'Integral' type power assisted steering unit, in which a cam and double-roller steering gear, the valves, cylinders and pistons are included. We now illustrate above the 'Universal' steering gear which incorporates the hydraulic control valves mounted upon our type '861' manual gear. This is for use with a separate power pump and with power cylinders operating on the steering linkage.

ADAMANT ENGINEERING CO. LTD., DALLOW ROAD, LUTON

Sole proprietors of the Marles Steering Company Ltd.

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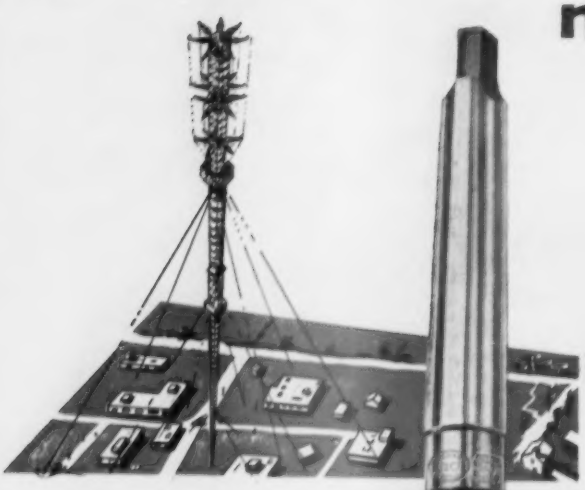
MARLES

An outstanding new feature



The best of two worlds is combined in 'TRUSHANK'—the latest refinement to the famous range of 'Mushet' brand high-speed steel twist drills. The precision ground taper shank ensures an accurate fit in the socket, whilst the body of the drill retains all the advantages of steam temper, with its distinctive blue-black oxide coating.

Order 'TRUSHANK' now and prove their greater efficiency during a longer and trouble-free life. They cost no more.

BRIGHT SHANK
Precision ground for accurate drilling.

STEAM TEMPER
Stress free, reduces friction and resists corrosion.

'DOUBLE MUSHET'
HIGH SPEED STEEL

'TRUSHANK'

TWIST DRILLS

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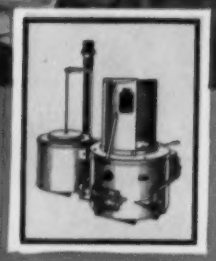
- *High output for moderate capital outlay*



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ICI

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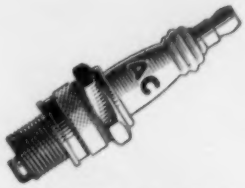
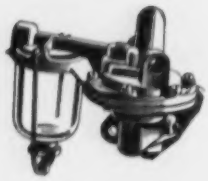
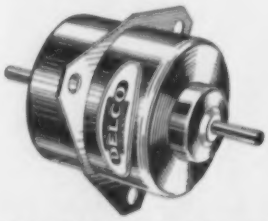


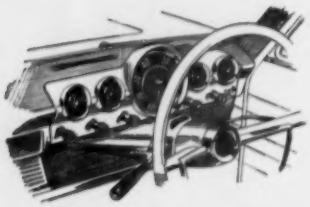
AC SPARK PLUGS



AC FUEL PUMPS



DELCO ELECTRIC MOTORS



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Virtually every British car has one or more AC-Delco components designed into it on the drawing-board. AC-Delco research engineers will gladly help when you are planning a new design or modifying an existing one. Liaison at the drawing-board stage can help your design and production programme.

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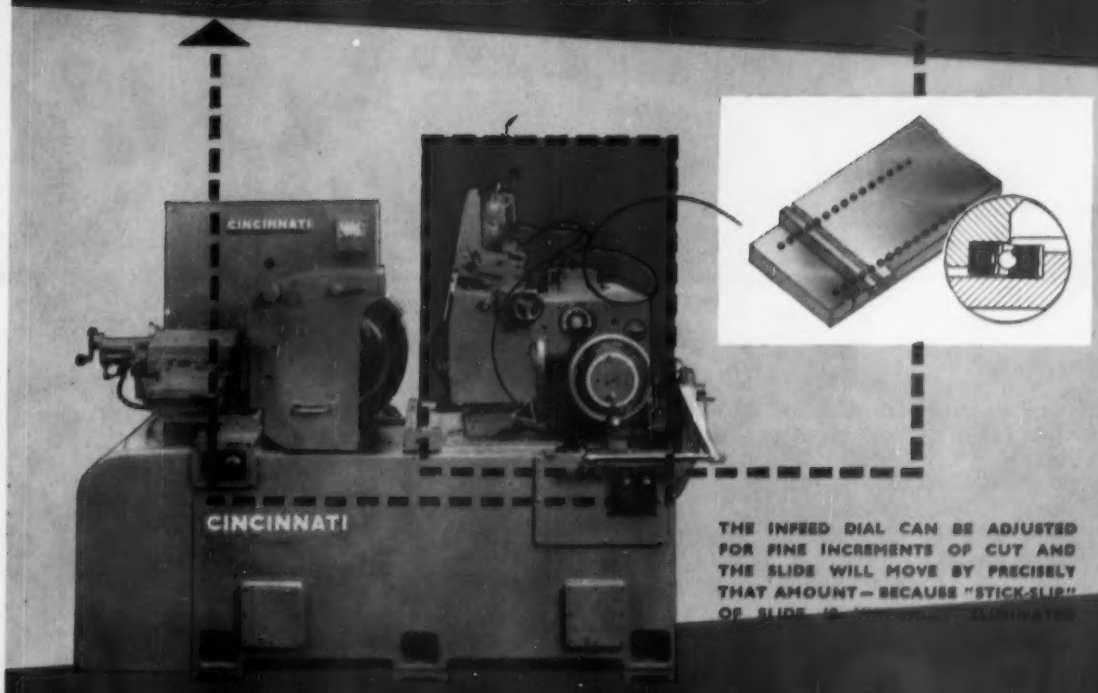

THE INFEED DIAL CAN BE ADJUSTED FOR FINE INCREMENTS OF CUT AND THE SLIDE WILL MOVE BY PRECISELY THAT AMOUNT — BECAUSE "STICK-SLIP" OF SLIDE IS

OTHER OUTSTANDING FEATURES OF THIS NEW

AUTOMATIC BALANCING OF THE GRINDING WHEEL

CINCINNATI MILLING MACHINES LIMITED
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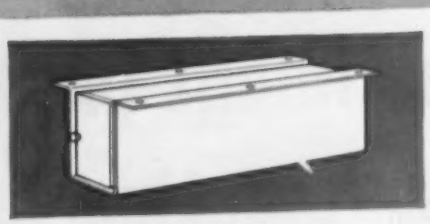
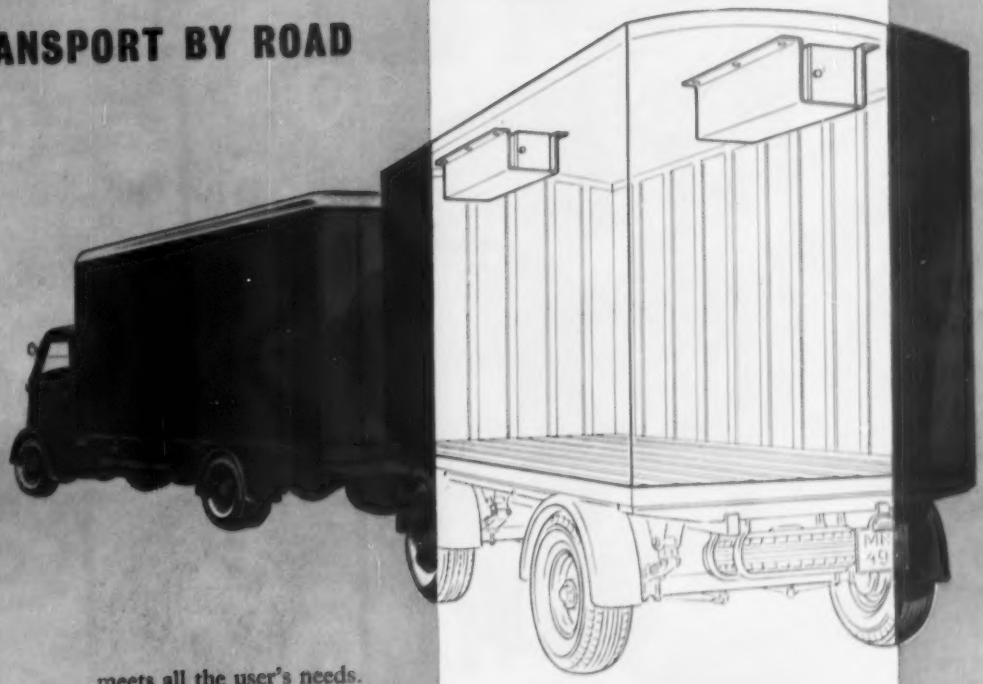
'DRIKOLD' REFRIGERATION FOR TRANSPORT BY ROAD





meets all the user's needs.
'DRIKOLD' does not reduce the payload ;
is simple and inexpensive ;
does not require skilled operation
or maintenance ;
is completely reliable in operation ;
is hygienic and free from mess.

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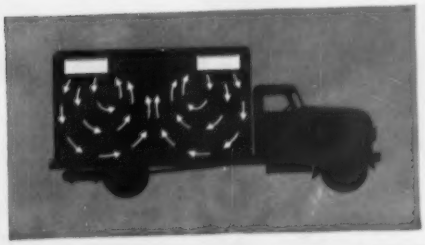

Before ordering or building Refrigerated Vehicles look into the outstanding advantages of 'DRIKOLD'

'DRIKOLD' **the solid carbon dioxide—dry ice—sold by Imperial Chemical Industries Ltd.**

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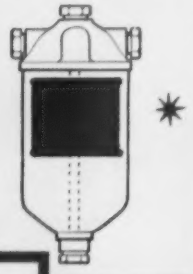
ICI

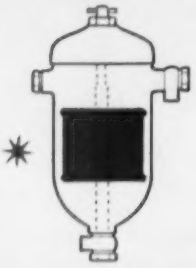
DR.330

Fit this NEW highly efficient PAPER element to your existing* C.A.V. fuel filters

Section of Cross Flow Filter, showing Paper Element Conversion.



Section of Down Flow Filter, showing Paper Element Conversion.



—Save on Diesel Maintenance. Conversion parts make fitting easy

So successful has the C.A.V. paper element filter proved that provision has been made for its benefits to be enjoyed by users of C.A.V. felt and cloth element filters; simple sets of conversion parts are available for models BFA 11 P, BFA 11 S, BFA 11 PS and BF 11 BS; fitting is a simple job taking only a few minutes, and the cost is low. In fact, conversion can save you money, apart from the saving due to greatly increased pump life.

Ask for full details. Conversion packs and replacement elements are obtainable from all C.A.V. service depots.

The new filter element can increase the life of your injection pumps by as much as eight times.

The World's Leading Manufacturers of

FUEL INJECTION EQUIPMENT

C.A.V. LIMITED, ACTON, LONDON, W.3

C.A.V.

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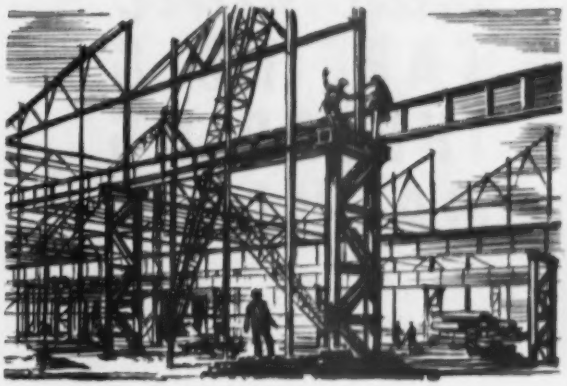

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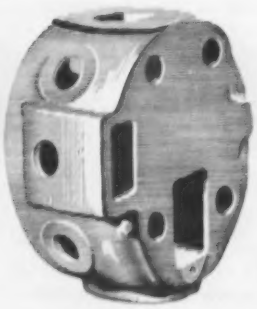
He's just going to use another of those

HARPER CASTINGS



28 lb. cylinder head casting for marine engine.

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Harper quality covers grey iron and Meehanite castings, also metal pressings, machining, enamelling and other finishes and sub-assembly work.

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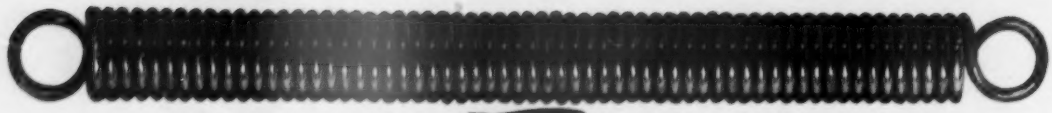
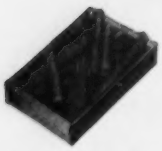
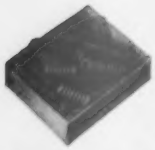
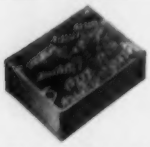




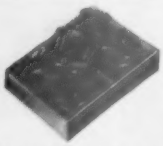
No. 760. 3 doz. Assorted Light Compression Springs 1″ to 4″ long, 22 to 18 S.W.G., ⅛″ to ½″ diam. 6/6 each.



No. 98A. 3 doz. Assorted 1″ to 4″ long, ¼″ to ½″ diam., 19G to 15G. 5/6 each.

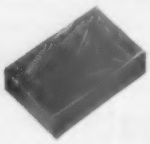


No. 757. Extra Light Compression, 1 gross assorted, ¼″ to ⅝″, ¼″ to 2″ long, 27 to 20 S.W.G. 15/- each.

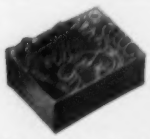


No. 388. ½ gross Assorted Small Expansion Springs. ⅜″ to 1½″, 18G to 21G. 9/6 each.

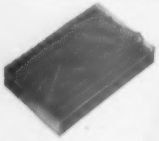
No. 758. Fine Expansion Springs. 1 gross Assorted ¼″ to ⅝″, ¼″ to 2″ long, 27 to 20 S.W.G. 15/- each.



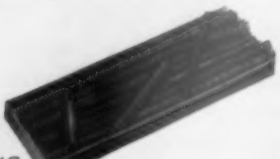
No. 466. ½ gross Assorted Small Expansion Springs ⅛″ to 1½″ long, 3/32″ to 3/16″ diam., 21G to 24G. 6/6 each.



No. 1013. 1 gross Small Coil Compression Springs, ⅜″ to 1½″ long, 3/32″ to 7/16″ diam., 24G to 19G. 6/- each.



No. 753. 3 doz. Assorted Light Expansion ⅛″ to ½″ diam., 2″ to 6″ long, 22 to 18 S.W.G. 10/6 each.



No. 1024. 20 Compression Springs 12″ long. ⅛″ to ½″ diam. 24G to 18G, suitable for cutting into shorter lengths; and 30 Expansions 1½″ to 12″ long, 5/32″ to ⅝″ diam., 22G to 16G. 24/- each.

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TERRY'S BOXES OF ASSORTED SPRINGS are just the job for your experimental department — a wonderful assortment of Compression and Expansion Springs . . . all sorts of lengths, gauges, diameters. The nine boxes we show are just a few from our range. Why not let us send you a full list—free?

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HT17(E)

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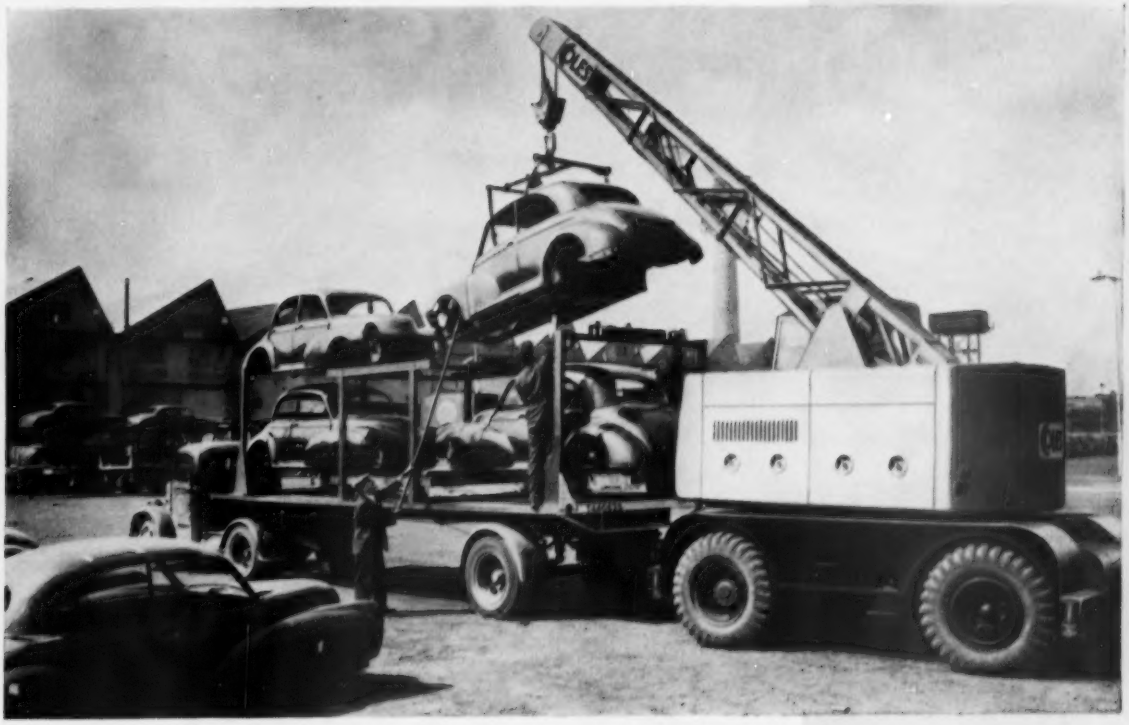
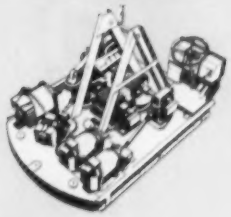


-of control and safety

Coles fatigue-free day long operation, pin-point accuracy and fast automatically safe motions, perform an unrivalled role in the automobile industry

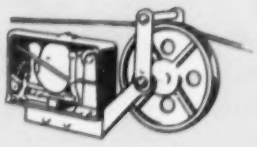
Essential for double-tier loading, Coles high angle cantilever jib and *Electric Transmission* system provide that extra margin of lift and safety with an instant choice of rapid or slow precise creeping speeds for lifting, derricking and slewing.

Always on hand to solve the numerous lifting problems that crop up in any factory, Coles have many saving ways.



ELECTRIC TRANSMISSION

Coles have cleared the decks - no inaccessible spots here. Individually powered electric motors provide smooth sustained power for all motions.



SAFE LOAD INDICATOR

Warns visually and audibly when maximum load is reached in any jib position, and positively cuts the motion if the warning is ignored

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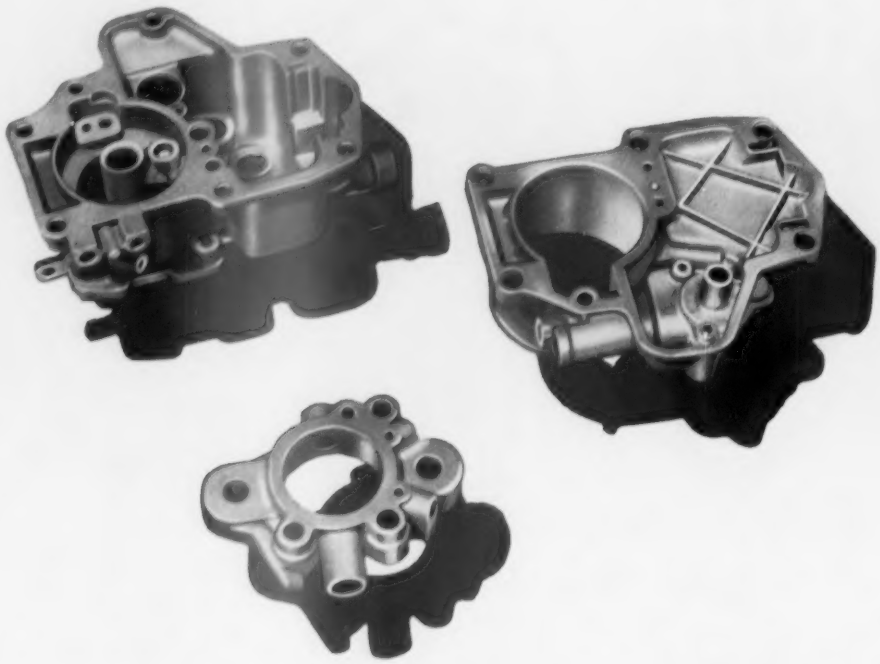
TODAY'S PROBLEMS IN INDUSTRY

Cutting Machining Time



In these days of competition both at home and abroad, machining time is a vital factor in maintaining competitive prices. It is expensive in labour and materials, and accounts for a considerable proportion of overall production costs. One way of cutting it to a minimum, however, is zinc die casting. Consider carburettors, for example. These are one of the first component parts for which zinc alloy die castings were regularly used, and as their design has become more and more complicated, die castings have proved themselves to increasing advantage. No other process allows such fast production of accurately held, complex shapes. Cored holes need only reaming or, where necessary, tapping; and the free machining qualities of zinc alloys make such operations easy and fast. Mating surfaces can often be used as cast. Justifiably this process is claimed to be the shortest distance between raw material and finished product.

Write for list of members and publications describing the properties and uses of die castings to:—

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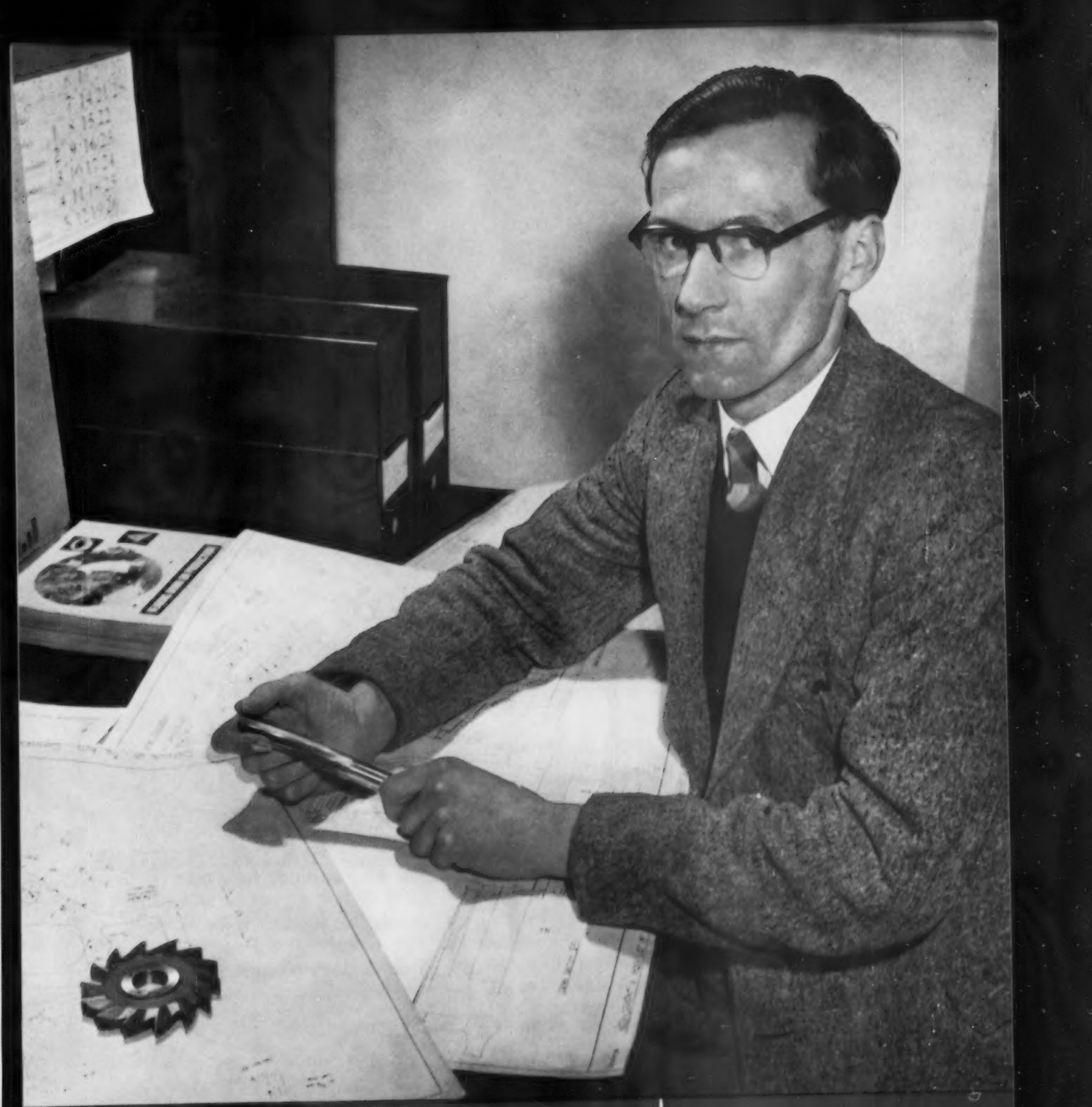
ZINC ALLOY DIE CASTERS ASSOCIATION

34 Berkeley Square, London, W.1. Telephone: GROsvenor 6636



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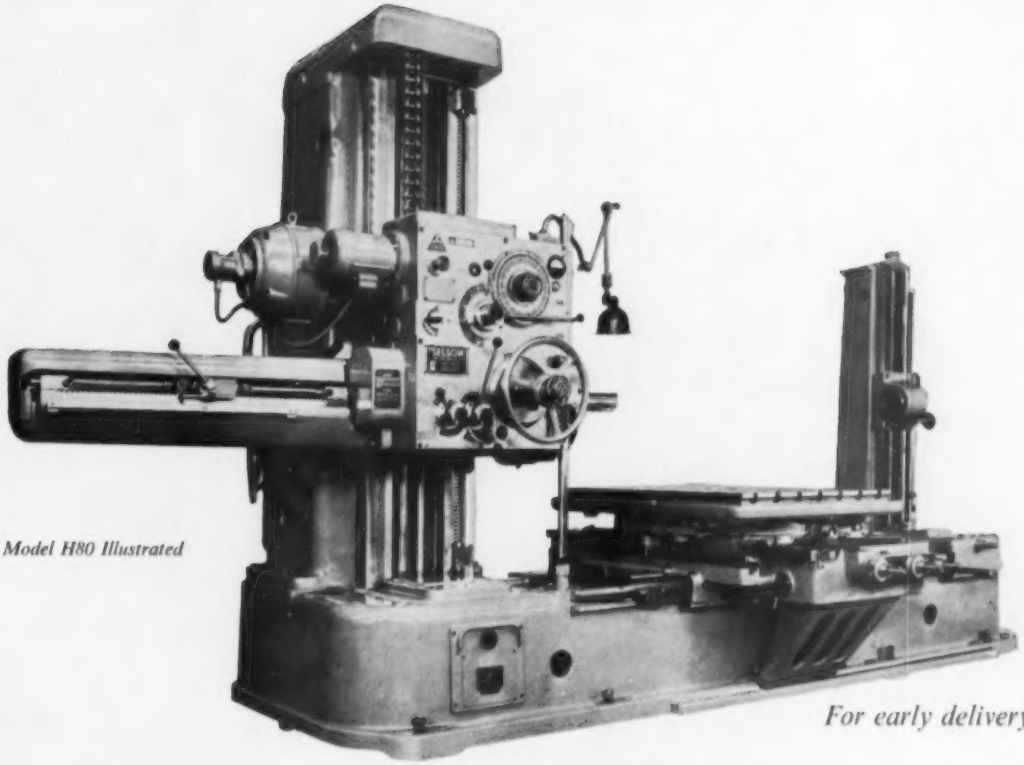

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HORIZONTAL BORING and FACING MACHINES

Safety Clutch disengages automatically working feed or rapid traverse to prevent damage. Power feed to table (Longitudinal, Cross and Circular) • Push Buttons allow fine adjustment of Spindle or Faceplate when setting • Automatic cross feed to facing head · Designed and built for Precision Boring, Drilling, Milling, Threading, Reaming and Facing · Enduring accuracy · High productive ability · Wide speed and feed range · Ease of operation.

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BRIEF SPECIFICATION

Model	H63	H80	H100
Diameter of Spindle	2½"	3½"	3⅛"
Max. diameter for facing	22"	28"	35½"
Working surface of table (width x length) ...	28" x 35½"	35½" x 44½"	44½" x 49½"
Number of spindle speeds	16	18	27
Range of spindle speeds: r.p.m.	8—1400	5.6—1000	2.8—710

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MODEL 3017—¼" DRILL

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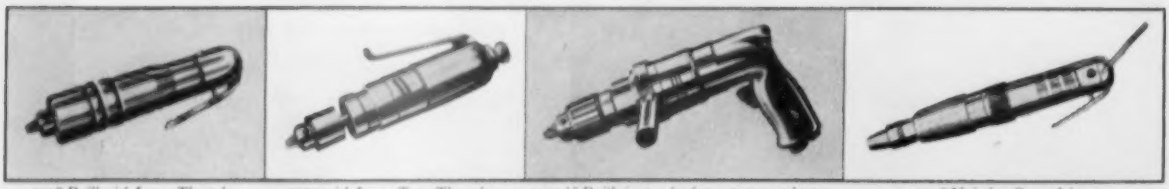
The line of drills ranges from $\frac{3}{16}$" to $\frac{1}{2}$" capacities in many speeds and a choice of lever or offset handles. The Screwdriver is the new supersonic machine in which noise is overcome by raising the exhaust above audio frequency.

All these tools are worthy of attention because of their high performance and complete reliability so important for production line work. Ask for descriptive literature.



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3008 Drills $\frac{1}{16}$"—¼" in speeds of 3,000 and 5,000 r.p.m.

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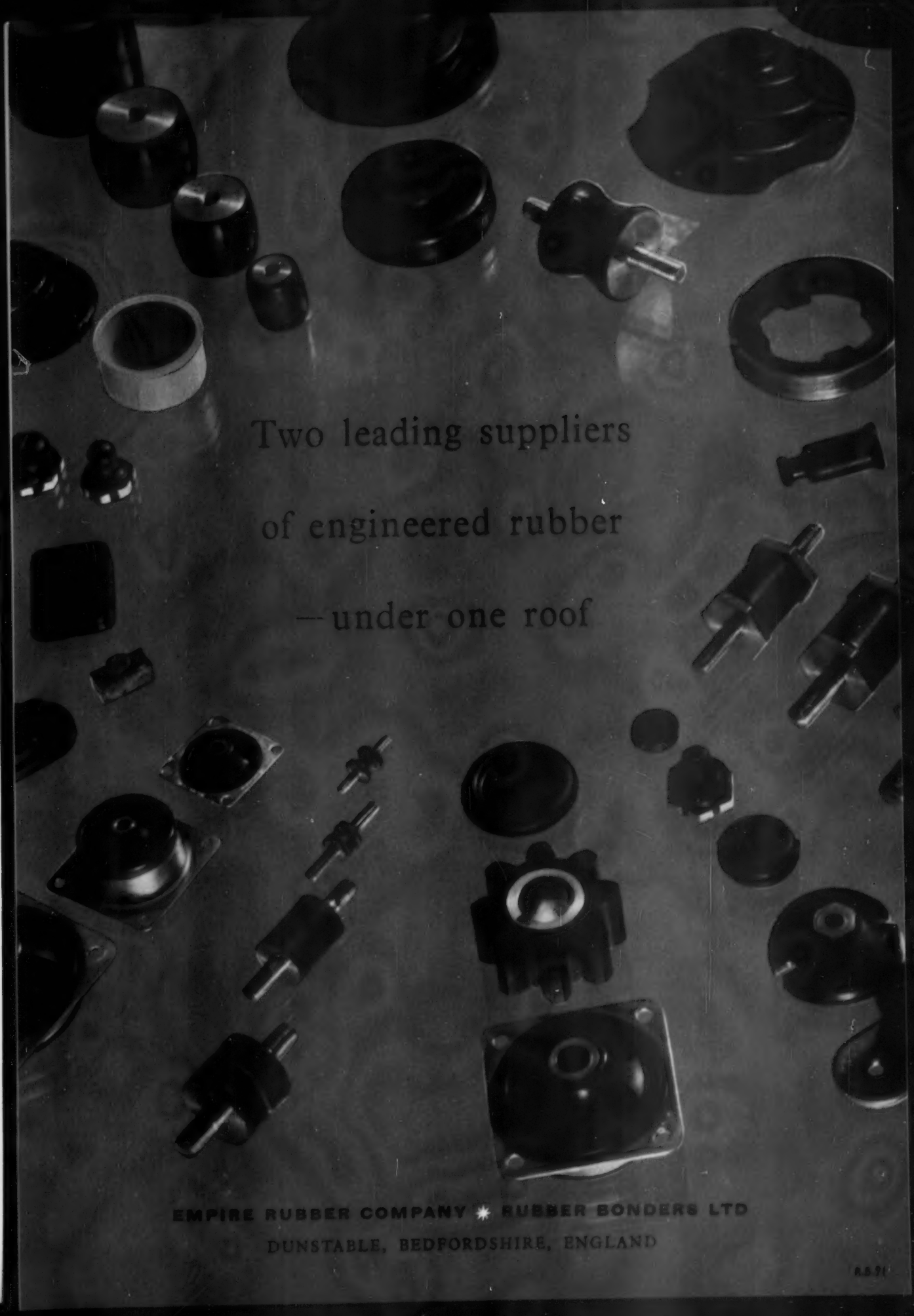
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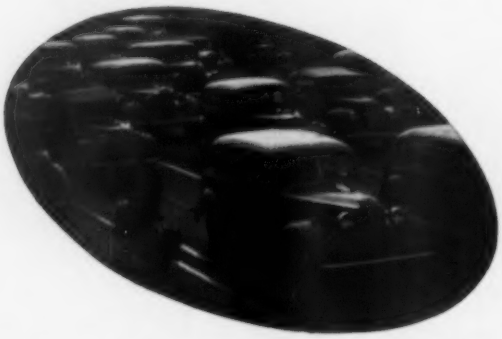

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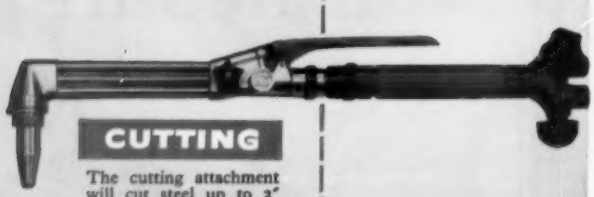
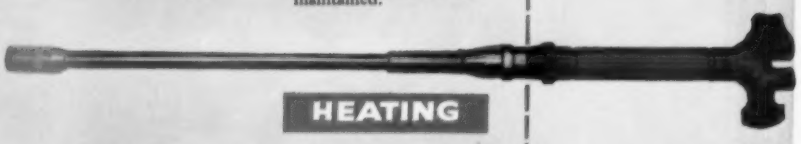


CUTTING

The cutting attachment will cut steel up to 2" thick. The highest degree of safety is ensured by the use of *nozzle-mixing*, limiting any possibility of backfire to the nozzle itself and allowing maximum cutting speeds to be maintained.



HEATING

The nozzles screw on to the straight welding neck or on to flexible necks which can be bent to any angle.

FLAME CLEANING

Flame cleaning attachments screw directly into the mixer and consist of a bent neck or a straight neck with a range of detachable flat or round nozzles.

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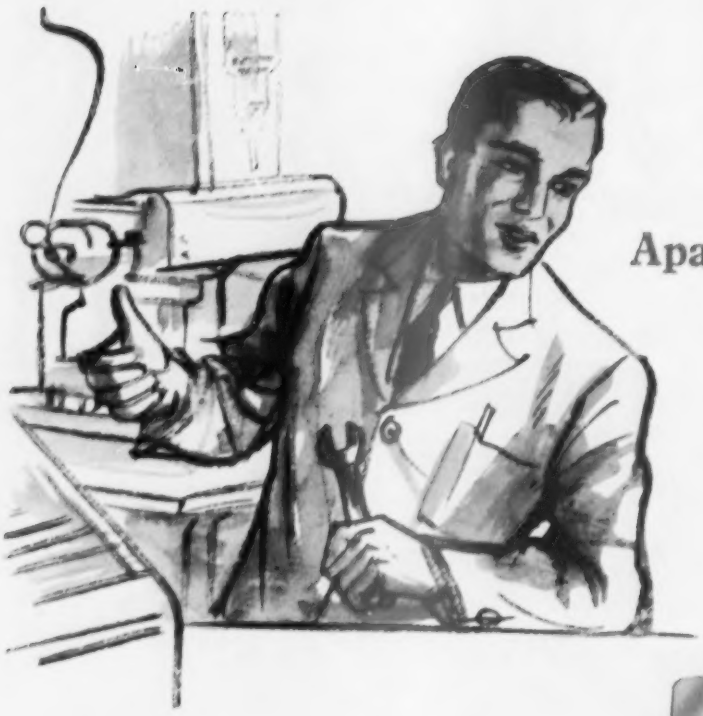

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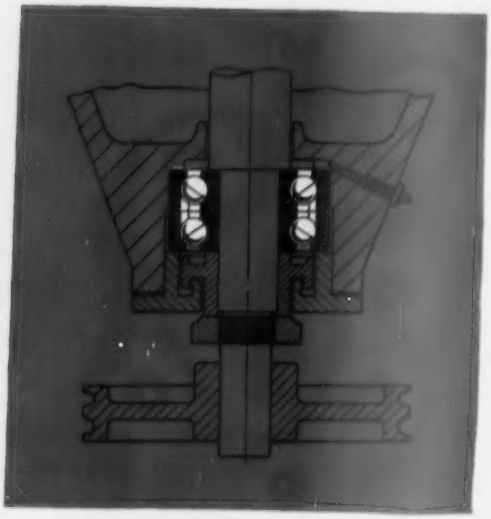

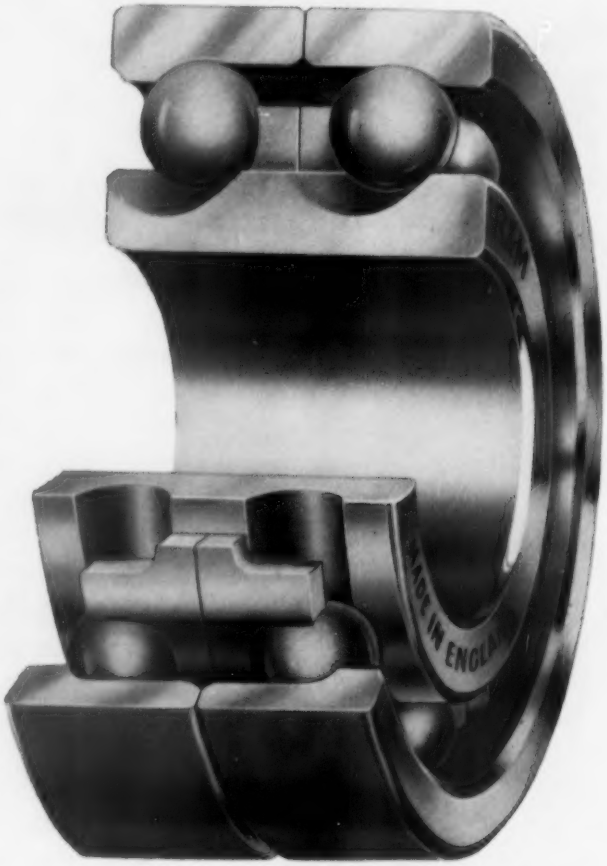

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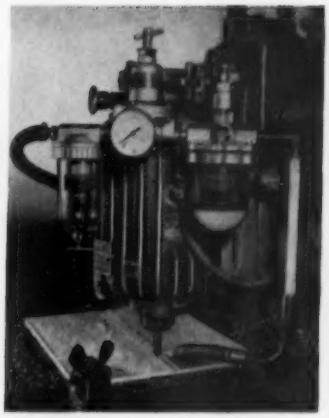

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Published the second Wednesday in every month by ILIFFE & SONS LIMITED
Dorset House, Stamford Street, London, S.E.1.
Telephone · Waterloo 3333 (60 lines)
Telegrams · Sliderule, Sedist London.
The annual subscription inland and overseas is £2 17s. 0d. including the special number Canada and U.S.A. $8·50

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G144

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F. C. Sheffield

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Commercial Vehicle Show

ALTHOUGH there have not been any revolutionary changes in commercial vehicle design since the 1954 Commercial Vehicle Show, trends demonstrating steady progress are nevertheless apparent. The work that is known to be going on behind the closed doors of the research and development departments of many manufacturers is unlikely to be shown until the new features have been well proved; in this respect British manufacturers differ from many of their foreign competitors.

The diesel engine continues to gain ground from the petrol type unit, and is now employed more widely than hitherto in light commercial vehicles, as well as in the heavier ones. With a view to reducing vehicle weight, further advances have been made in structural design and, increasing attention is being paid to the use of laminated plastics for cab and body construction. An important advantage of plastics is that for relatively small quantity production the tooling costs are much lower than for steel pressings. Furthermore, panels damaged in service can be repaired relatively easily. All these features reflect the increasing economic pressure on operators and the consequent need to reduce operating costs to an absolute minimum.

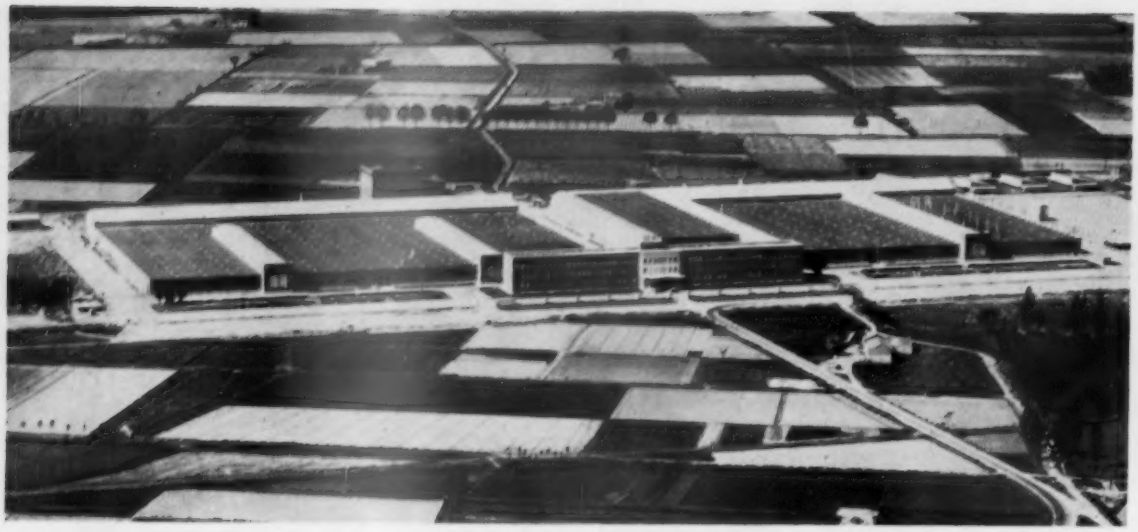
Expanding business in export markets, where very long-distance hauls are common, has led to further efforts to ease the task of the operating crews. Progress has been made in three main directions: greater comfort is offered by the incorporation of better seating arrangements, noise reduction, cab ventilation and temperature regulation; two-pedal control makes for easier driving in heavy traffic; and, thirdly, more vehicles are designed to incorporate power-assisted steering, the demand for which is definitely increasing because of the extra weight carried and the larger tyre sections now employed as a result of changes in the British legal requirements for heavy vehicles.

The changes in the British regulations limiting overall dimensions and maximum weights have helped both manufacturers and operators. These new regulations are almost in line with those in force in most foreign countries, so manufacturers have been able not only to effect a greater degree of rationalization in their vehicles for the home and overseas markets, but they can now also compete on more favourable terms in some countries where hitherto the British restrictions with regard to size and weight placed them at a serious disadvantage in that they had to make special vehicles for export. Operators, of course, benefit from reduced prime costs, resulting from rationalization, and also because increased load carrying capacity tends to reduce operating costs.

In general, the increases in size and weight have led to the strengthening of frames, adjustment of spring rates and improvements in braking. The larger section tyres have not been difficult to accommodate because of the increase allowed in overall width of the vehicle.

The evolution of new light-weight vehicles designed specifically for commercial applications is a welcome feature. They are in almost every respect a marked improvement on the earlier adaptations of passenger car designs. Provided the basic design of these light commercial vehicles is suitable for a wide range of models, such as vans, pick-ups and passenger carriers of different types, including ambulances, there is ample justification for investment in tooling for production. Although there is a divergence of opinion as to whether or not a separate frame should be employed, the steady incursion of integral construction into the field of commercial vehicle design continues, and it seems unlikely that exceptions will be made in respect of the light-weight types. In fact, the need for a low floor level will probably be widely regarded as a deciding factor in favour of integral construction.

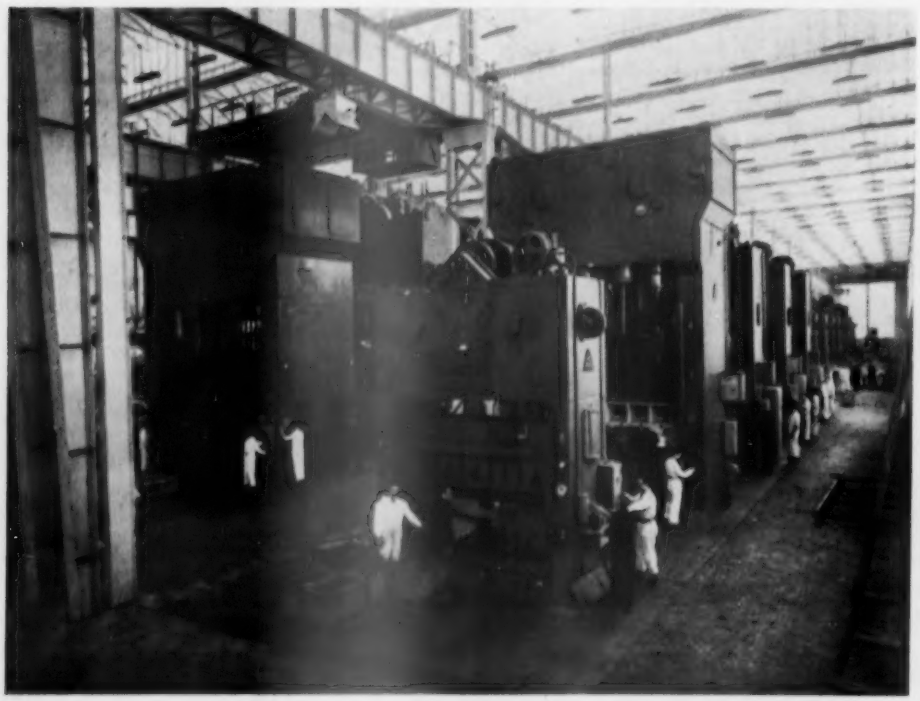
Automatic and semi-automatic transmissions are being introduced for private cars, and probably most will be offered also for light commercial vehicles. A large proportion of these vehicles operate in towns, and it is in heavy traffic that maximum benefit is obtained so far as reduction of driver fatigue is concerned. Obviously the systems that do not increase fuel consumption will be most sought after by operators.



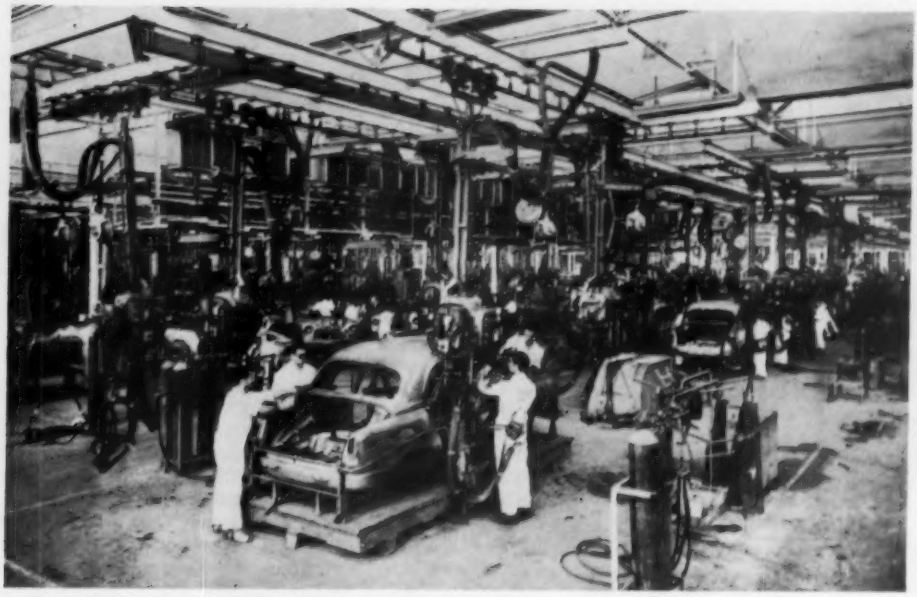
Spanish Automobile Production

The Plant of S.E.A.T. at Barcelona

IT is the eventual intention that the new works of the Sociedad Española de Autómoviles de Turismo shall render Spain self-supporting so far as the smaller and relatively inexpensive passenger cars are concerned. The plant, which is in the Free Zone of Barcelona, was officially opened at the end of 1955. At present it employs 1900 workers. It is predominantly a government enterprise, although part of the capital of 600,000,000 pesetas (about £5,500,000) was subscribed by six leading Spanish banks and by the Italian Fiat company. The participation of the Fiat concern is not only a consequence of the fact the Fiat 1400 is the vehicle selected for Spanish manufacture, but is also official recognition of the efforts made by Fiat since 1930 to establish car manufacture in Spain.



The heavy press section

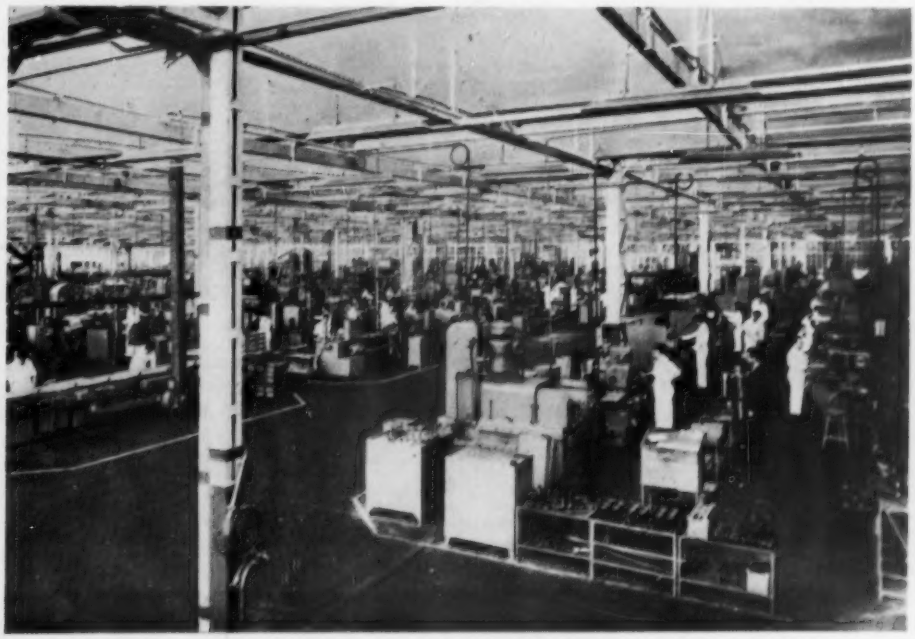


Building Fiat 1400 bodies in the SEAT works

An annual production of 10,000 Fiat 1400 cars is planned. In addition, it is planned to undertake in 1957 the manufacture of Fiat 600 cars, for which the production target is 20,000 per annum. Of the initial capital, more than half was spent in the purchase of machine tools from the United States of America and the five main European manufacturing countries. Incidentally, since the inauguration of this concern the capital has been increased by 50 per cent to 900,000,000 pesetas.

The SEAT factory has a productive floor area of 900,000 square yards. There is evidence of Fiat influence in the machines employed and in the general lay-out of the plant. In the machine shop the machines are arranged transversely in lines to give flow production through all the machining operations, to heat treatment, if it is necessary, and thence to the sub-assembly and final assembly lines. In all there are 31 lines, incorporating 104 lathes, 47 millers, 6 broaching machines, 42 grinders, 23 automatic screw machines, 21 gear cutters, 6 hypoid generators, 7 gear testing and finishing machines, 4 gear shavers and acoustic testers, and other classes of machines. In the heat-treatment section there are 17 furnaces of the latest type, including salt baths and gas cementation furnaces. There are two degreasing plants in this section, and provision is also made for shot-peening.

Engine testing is carried out to a degree that is no longer considered necessary in this country, at least for engines of the type and capacity produced by SEAT. The engine test section has eight test beds, equipped with variable speed motors to which the engines are coupled. Testing is effected in two stages. In the first stage, which lasts an hour, the



A portion of the well-equipped machine shop

The final assembly line

motor drives the engine, while in the second stage, which lasts 15 minutes, the engine drives the motor as a generator at speeds of 500, 1,000, 1,500 and 2,000 r.p.m. Also included in the equipment of the engine test section is a Froude brake for more extensive tests on a percentage of the engines produced.

In the press shop there are 25 presses of 250–1500 tons capacity, 31 medium presses of 59–200 tons capacity and 25 mechanical presses of 8–55 tons capacity. The work is put through in batches, each sufficient for two to three months requirements. Incidentally, the press shop capacity is in excess of present requirements. Although the actual presses are of the most modern types, apparently there has been no attempt to expedite production by the use of mechanical handling devices such as iron hands.

From the press shop the work is passed to an intermediate store to be held until required for assembly. From the stores, the work, mounted on suitable fixtures, is passed as required to eight sub-assembly lines. The plant employed for sub-assemblies comprises 67 electric spot welders, nine arc welders, one continuous and seven autogenous welders. Completed sub-assemblies are mounted on a special fixture for welding to form the body shell. The illustration of the body shell section shows that the methods adopted in the Fiat factory at Turin have also been adopted in great measure in this factory. The body shell is transferred to the final welding and finishing line on a travelling belt.

On leaving the finishing line, the body shell is transferred by overhead monorail to the painting section. There are two main conveyors in the painting section. On the first of these the shell passes through the following sequence of operations: degreasing and de-rusting, slight phosphating, washing, the application of anti-noise paste and polishing. During its passage on the second conveyor, three coats of cellulose are applied, the shell is oven baked and is given a final polishing. Apart from the two main conveyors in the painting section there is also an auxiliary line served by overhead transporters for the painting of wheels and loose parts. The section is air-conditioned and pressurised to prevent the entry of air from outside.

Finished shells are transferred to a conveyor, 420 ft long, for the attachment of the petrol tank, windscreen wiping mechanism, electric wiring, door latches, etc. The trim section is situated alongside this conveyor. The remainder of the vehicle assembly is carried out in accordance with what is standard practice in most automobile factories in this country and on the Continent.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

OCTOBER

London

Tuesday, 9th October, 6.0 p.m., at 1 Birdcage Walk, Westminster, S.W.1. Annual General Meeting followed by Ordinary General Meeting. Chairman's address: "Experiences During Forty Years of Automobile Design", by A. G. Booth, M.B.E., M.I.Mech.E.

Birmingham

Tuesday, 30th October, 6.30 p.m. in the James Watt Memorial Institute, Great Charles Street, Birmingham. Paper: "Proof Testing of Electrical Equipment", by R. F. Groves, B.Sc.(Eng.), A.M.I.Mech.E.

Derby

Wednesday, 17th October, 7.0 p.m. in the Midland Hotel, Derby. Paper: "An Experimental Turbo-Diesel Locomotive," by F. L. Picard. Presented by R. W. F. Mitchell and R. Vuaillat.

Luton

Monday, 8th October, 7.30 p.m. in the Assembly Room, Luton Town Hall. Address by the Chairman of the Centre, A. W. Larking: "Contributions to Engine Life".

North-Eastern

Wednesday, 10th October, 7.30 p.m. in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Front Suspension and Tyre Wear", by V. E. Gough, A.M.I.Mech.E. and G. R. Shearer, G.I.Mech.E.

North-Western

Tuesday, 9th October, 7.15 p.m. in the Grosvenor Hotel, Chester. Paper: "The Effect of Maintenance on Injection System Performance", by W. A. Mason (Shell Research Ltd.).

Scottish

Monday, 15th October, 7.30 p.m. in the Institution of Engineers and Shipbuilders, 39 Elmbank Crescent, Glasgow, C.2. Chairman's address: "Experiences During Forty Years of Automobile Design", by A. G. Booth, M.B.E., M.I.Mech.E.

Friday, 19th October, 7.30 p.m. in the Robert Gordon Technical College, Aberdeen. Paper: "Sales and Service in the Transport Industry", by J. M. Forbes.

Western

Tuesday, 30th October, 7.30 p.m. in the Royal Hotel, Bristol. Chairman's address: "Experiences During Forty Years of Automobile Design", by A. G. Booth, M.B.E., M.I.Mech.E.

NOVEMBER

Coventry

Tuesday, 6th November, 7.15 p.m. in the Grosvenor Room, Leofric Hotel, Coventry. Chairman's address: "Experiences During Forty Years of Automobile Design", by A. G. Booth, M.B.E., M.I.Mech.E.

FREE-PISTON ENGINES

A Development Programme for Automotive Units by the Free Piston Engine Co. Ltd.

In March of this year, Associated British Engineering Ltd. signed two agreements with Alan Muntz and Co. Ltd. The first concerned Pescara free-piston gasifiers having diesel pistons of 6in diameter and over, and covered the manufacturing and selling of gasifiers in the United Kingdom and British Commonwealth on a non-exclusive basis. Two such gasifiers are already designed and in production, namely the S.I.G.M.A. GS34 model of 1,250 gas h.p. and the Muntz CS75 model of 420 gas h.p.

The second agreement concerned Pescara free-piston gasifiers having diesel pistons of under 6in diameter, and covered the manufacturing and selling of gasifiers in the United Kingdom and British Commonwealth on an exclusive basis. There are no such gasifiers currently available for production. Two exist in prototype development stage, the Renault G.12 unit of 132/162 gas h.p. and the General Motors 4-4 Hyprex engine of 250 gas h.p.*

The fields of application envisaged for this "under six inch" free-piston engine are road transport, tracked vehicles, earth-moving equipment, rail cars and high-speed marine craft. A subsidiary company, Free Piston Engine Co. Ltd., is responsible for the design, development and sales, while manufacture, assembly and test are being undertaken by a Free Piston Division of Henry Meadows Ltd.

A design study is under way to determine the optimum horse-power of the new gasifiers and the consequent leading dimensions. Taking 250 gas h.p. as the power requirement, the following specification was obtained:

Engine cylinder bore	5·78in
Compressor cylinder bore	16·00in
Overall length	5·79 ft
Continuous maximum rating	250 gas h.p.
Stroke at continuous max.	7·32in per piston
Oscillations at continuous max.	1,300 c/min
Weight	2,290 lb
Power:weight ratio	9·17 lb/g.h.p.

Whilst 250 g.h.p. in one unit would be satisfactory for numerous applications, it would be excessive for others. An alternative machine of 125 g.h.p. has, therefore, been investigated. The first assessment has given the following specification:

Engine cylinder bore	4·07in
Compressor cylinder bore	11·29in
Overall length	4·07 ft
Continuous maximum rating	125 g.h.p.
Stroke at continuous max.	5·16in per piston
Oscillations at continuous max.	1,840 c/min
Weight	811 lb
Power:weight ratio	6·5 lb/g.h.p.

This dimensional analysis is based on the CS75 machine and no attempt was made to achieve anything in the nature of a lightweight design. The expected change in performance of the 125 g.h.p. unit upon substitution of light moving parts would be:

Continuous maximum rating	153 g.h.p.
Oscillations at continuous max.	2,400 c/min
Weight	618 lb
Power:weight ratio	4·03 lb/g.h.p.

Since this is in excess of the required output in some instances, a further reduction in size becomes permissible and gives the following specification:

Engine cylinder bore	3·66in
Compressor cylinder bore	10·15in
Overall length	3·66 ft
Continuous maximum rating	125 g.h.p.
Stroke at continuous max.	4·65in per piston
Oscillations at continuous max.	2,640 c/min
Weight	452 lb
Power:weight ratio	3·62 lb/g.h.p.

It should be understood that these are first and approximate assessments. The ultimate gasifier may be a somewhat different machine when the many problems of detail design have been solved. The target for completion of a prototype machine is eighteen months, with probably a further eighteen months for prototype testing.

An outstanding characteristic of the free-piston engine is its ability to run on virtually any of the liquid petroleum hydrocarbons. In France, S.I.G.M.A. have reported running thousands of hours on residual fuels with sulphur contents up to 4 per cent. General Motors have successfully operated their unit on 100 octane gasoline. The practical range for automobile purposes would appear to be from high octane gasoline to marine diesel fuel.

* "Automobile Engineer" August 1956

COMPARISON OF ENGINE TYPES

Type	Fuel Consumption Lb/bhp/hr		Weight Lb/hp	Size ft³/100 hp	Fuel
	maximum power	optimum economy			
Spark ignition piston engine, Cast iron, 9·1 Comp. ratio	0·48	0·41	3·6	9·1	95 Octane
Spark ignition piston engine, Aluminium, 8·1 Comp. ratio, Turbo-charged—30 per cent boost	0·49	0·43	2·1	7·3	95 Octane
Gas turbine, Regenerative	0·75	0·75	3·6	7·2	Kerosine
Diesel engine, 4-stroke cycle	0·41–0·47	0·38	12·7	26·0	No. 2 Diesel
Diesel engine, 2-stroke cycle	0·42	0·40	12·1	14·8	Diesel
Diesel engine, 2-stroke cycle, Exhaust turbo-blower	0·42	0·41	11·7	13·6	Diesel
Free-piston engine	0·48	0·48	3·6	7·0	Kerosine Gasoline Diesel

Engine Installation

A Discussion of Methods to Suit Unitary Construction

H. C. Harrison, A.R.C.S.

AMONGST the first decisions to be made in the design stages of a new vehicle is the cylinder configuration of the engine. The decision is a weighty one and, of all the factors to be considered, it is difficult to determine which shall be given preponderance over another. In the course of development down the years this country has become largely a four cylinder country just as the United States is an eight cylinder country. And where immense investments in machine tools and equipment are consolidated in the production of a particular type, those circumstances may, indeed, determine the policy to continue with it against the strongest technical representation for any change. Further the four cylinder engine has given such worthy service over the years and has become standardised at such a high level of performance and reliability that its position is hard to challenge. Whilst that development has been going on, however, refinement has also progressed in other directions—transmission and suspension, and the rather revolutionary unitary body construction. Such advances call for that "second look" at the engine to check that it is not getting left behind in the general progress. This present review is directed to that purpose. The superior class of goods, motor cars included, will not be denied their market,—a fact well appreciated by competitors overseas.

Historical

A brief historical survey is appropriate. As a starting point we may reach back to the day when an engine of some 8 or 10 brake horse-power was regarded as a fairly powerful unit to install in a vehicle fitted with wooden wheels and solid rubber tyres. The imperfect balance of the two cylinder engine (or even single cylinder) was obscured by other disabilities of the vehicle as a whole.

As the vehicle emerged from the horseless carriage category, it came to be realised that its engine must possess inherent balance; at the same time advances in other directions—design, materials, and manufacturing technique, made the multi-cylinder engine practicable. In consequence the four and six cylinder engines became established at quite an early date and have remained conventional power units in this country ever since, in spite of spasmodic challenges at various times from straight 8's, V8's and some few other combinations.

Each version had its own protagonists. Those who favoured the six advanced the opinion that since it was mechanical balance that was sought, the only logical answer was a design that offered complete balance and this was to be found in the six cylinder arrangement. No alternative arrangement could possibly offer anything in advance. A benign Providence had provided everything in one package. Both fundamental and second order harmonics were completely balanced, as were couples of the first and second order. Harmonics of a higher order were so small in magnitude as to be insignificant and could indeed be overshadowed by necessary manufacturing tolerances. True there remained slight imperfections causing oscillations in roll, yaw, and pitch, arising from the recurrent firing impulses, the inertia of reciprocating masses, and lack of uniformity of mixture distribution to the cylinders. Obviously these imperfections had to be tolerated until the arrival of some form of turbine prime mover,—a very long vision for the crystal-gazer of the period.

Homage to the gift of providence was paid by one manufacturing company (the Napier Company as then constituted). They were determined that nothing should jeopardise the inherent balance of the six. At a period when precision forgings were still in the future and the dynamic balancing machine was only just appearing on the horizon, the company produced six-throw crankshafts on the lathe from a solid billet. Crank-webs were formed as circular discs of the diameter of the billet,—the journals and throws being carved out of the solid metal between them. Geometry, of itself, took care of the balance at a cost they accepted as the price of perfection.

Those who were not prepared to swing over to the six found a great deal of merit in the four. It was in primary balance and the firing intervals were evenly spaced,—two attributes which were absent from earlier engines but which had now come to be regarded as an essential minimum. Complete inertia balance was, of course, greatly to be desired but it could only be attained at the cost and complexity of two additional cylinders and their related parts. But, again, the standard of engineering refinement of the rest of the vehicle was such that the lack of complete balance could be accepted.

At about this time Dr. F. W. Lanchester developed two most ingenious mechanisms. One was the crankshaft torsional vibration damper which was particularly applicable to the six with its slender whippy crankshaft. Its introduction raised that engine to a new pinnacle of refinement by damping-down the elastic torsional oscillations induced by successive explosions, which could reach destructive proportions when they came into resonance with the crankshaft natural frequency.

The other invention was the harmonic balancer applicable solely to the four. The problem was to nullify the vibrations arising from the unbalanced 2nd order harmonic. The virtue of the device may be the better appreciated if we regard the engine as possessed of four pistons moving up and down in the cylinders in perfect harmonic balance plus a fifth phantom piston of approximately the same weight as one of the real pistons but totally unbalanced and driven by a phantom crankshaft-throw rotating at twice real crankshaft speed. The harmonic balancer neutralises the phantom piston by introducing vertical forces that are at all times equal and opposite to those imposed by the phantom. By so doing it "converts a four into a six" at insignificant cost as compared with the alternative of adding two additional cylinders and without the penalty of any serious increase in overall dimensions.

Each of these inventions was a unique solution to its problem. They were, indeed, born into an exhilarating atmosphere when improvement in engine design was taking place at a rapid rate. Established conventions had been severely shaken by the introduction of the Silent Knight double sleeve-valve engine and an accompanying spate of inventions for alternative valve arrangements. Designers were compelled to refine their cam-operated poppet valves and so stave off a strong threat of submergence by the newcomer. All this striving for improvement opened the

field in the sense that many a device, which would previously have been brushed aside as an unnecessary complication, was now considered more closely. As a competitor of contemporary poppet-valve engines with exposed tappet and rocket gear, the Knight was justly claimed to have the attribute of silence and moreover it confounded the critics who prophesied complete breakdown of cooling and lubrication. Various models were exported in some quantity to U.S.A. and the Continent. In the course of a famous endurance test conducted under R.A.C. supervision a batch of some six or eight engines established an undreamt-of reputation for reliability by running on the test bed under full power night and day for a week. In retrospect it can be seen that in its day the Knight engine punctuated the progression of automobile design just as unitary body construction has done at this later date. Much more was subsequently demanded of the engine as a power unit, and in turn attention was directed to the transmission to bring it into line with the new standards set by the sleeve-valve engine.

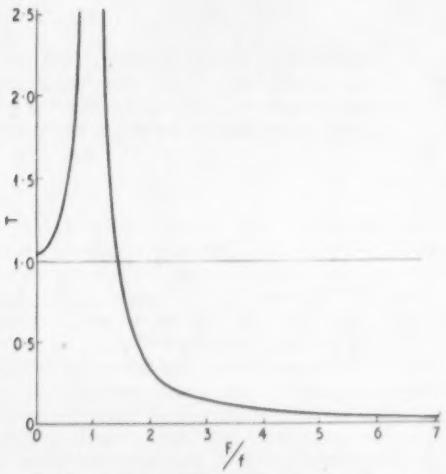


Fig. 1. Transmissability of vibration frequencies

Coming to the installation of the engine in the vehicle the aim at first was to simulate the installation of a stationary engine securely bolted to its massive bed which supposedly, by reason of its inertia, absorbed all vibrations. Chassis frames were rugged and, in some degree, acted in this manner. A sub-frame between engine and chassis was later introduced providing the opportunity for fitting resilient packings at the point of attachment and so absorbing some of the vibration. Further insulation was introduced at body mounting points. These practices were of course applicable equally to both four and six but they enabled the four to maintain its popularity in spite of the inherent handicap of inferior balance as compared with the six.

Engine vibration

With the development of unitary body construction the position has undergone a substantial change. In the absence of a chassis frame and any insultaion between it and the body, the engine mounting has assumed tremendously greater importance particularly in the case of the four. Engine vibrations transmitted into the body can have the effect of setting the panels into a state of resonance in sympathy with the engine vibrations so that they emit a sustained musical note commonly referred to as "boom". The noise may reach intolerable proportions and in any case is entirely out of accord with other standards of comfort built into a high quality vehicle. Sound-deadening the panels by means of flock-spray and similar expedients is generally ineffective; the real remedy is to be sought at source by close attention to the engine mounting.

Two phases of the problem are recognised: In the first place, since the engine gearbox unit is in a state of vibration from the causes already mentioned, means must be found to prevent these vibrations from being transmitted to the body. Secondly, there are harmonic alterations in the value of the torque as transmitted from the engine to the propellor shaft which are exhibited as an harmonic variation in the reaction forces in the rear axle and which from thence are transmitted to the springs. The usual insulation in this case takes the form of rubber bushes fitted to the spring eyes, supplemented sometimes by rubber pads at the spring-seat on the axle.

Regarding the first mentioned phase—that of vibration of the engine mass—it is convenient here to refer to the text-book diagram illustrating transmissability of vibration frequencies (Fig. 1). Whatever the form of the engine mounting, it will be possessed of a natural frequency of vibration (f)—the frequency at which the engine whilst stationary, upon being displaced from the position of rest and then left to itself, will vibrate until it comes to rest again. When the engine is running its own vibration frequency (F) is superimposed on the natural frequency of the mounting and is transmitted in greater or less degree through the mounting to the supports. In Fig. 1 the ordinate T is the transmissability—the proportion of the applied force (or amplitude of vibration) which is transmitted through the engine mounting, the abscissa is the ratio F/f: The expression connecting these quantities is

$$T = \frac{1}{\left(\frac{F}{f}\right)^2 - 1}$$

It is evident that when F/f is unity, the system is in resonance and the transmitted amplitude becomes as large as permitted by the physical proportions of the mounting. At the value $\sqrt{2}$ for the ration F/f, the transmitted amplitude is equal to the applied amplitude whilst for higher values it falls off rapidly until at about a ratio of six or seven the engine may be deemed almost stationary.

Fig. 1 does not take into account any damping co-efficient as is indicated by the fact that the value of T inclines to infinity when the value of F/f is unity. But in practice some degree of damping is present in any form of mounting; a higher value for the ratio F/f is required than is necessary without damping. The object therefore in designing a mounting is to achieve a low natural frequency of the engine on its mountings. That frequency is not easy of direct measurement but it may be readily computed from the expression

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{M}}$$

vibrations per sec in which "k" is the stiffness of the mounting element measured as lb per inch of static deflection and "M" is the mass (slugs) of the supported weight. Both of these quantities can be easily determined. For a particular mass therefore a low value of "f" is given by a low value of "k". This calls for a "soft" mounting and some difficulty arises in reconciling this requirement with the need for providing adequate resistance against road bumps when travelling.

A similar expression applies to a mounting called upon to exert a couple for resisting torsional oscillations. Here the natural frequency is given by

$$f = \cdot 159\sqrt{\frac{q}{MI}}$$

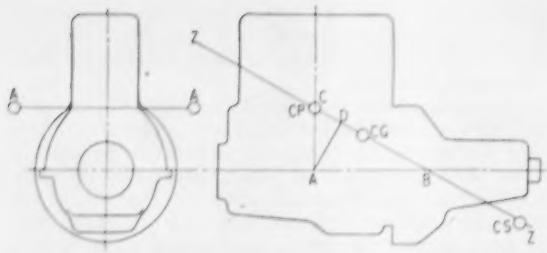


Fig. 2. Essential requirements for mounting a four-cylinder engine

cycles per sec where q is the stiffness of the mountings measured as lb-in per radian deflection and MI is the moment of inertia about an axis normal to the plane containing the couple.

Now in a four cylinder engine the 2nd order harmonic (and likewise the torque harmonic arising from the firing impulse) has a frequency of twice engine speed. The numerical value of the force it exerts increases as the square of the speed. At low engine speeds the frequency is below that which is audible to the human ear, but as audible frequency is approached so also does the energy of the vibrations increase until at a frequency of about 64 cycles per second (two octaves below middle C and usually corresponding to a road speed in the region of 30 miles per hour), the energy content of the vibrations is sufficient to force them past the damping and so bring into resonance any panel which itself possesses that frequency as a natural period of vibration. The panel then emits "boom".

Engine mounting

The essential requirements for mounting the engine in the vehicle are relatively simple. They are considered here in relation to a four cylinder engine, Fig. 2. All vibrations are referred to a principle axis ZZ which passes through the centre of gravity of the engine-gearbox unit and about which axis its moment of inertia is least. If the unit were permitted a freedom of oscillation about any other axis, lateral forces would be introduced at the mountings and their accommodation would lead to undue complication. This theorem, as well as the location of the principal axis and the derivation of the moments of inertia referred to it, are the subject of text-book propositions. (See also article entitled "Engine Mounting" in March, 1953 issue of this journal). The advantages to be gained from making use of the properties of the principal axis are well recognised but they have not always been fully exploited.

We seek first to accommodate the forces arising from the 2nd order harmonic, these being the only translational forces called into play. They occur as vertical forces in the transverse plane midway between the cylinders. They could be ideally resisted by a mounting situated within the body of the engine directly in line with the applied force but since this is obviously impossible, two mountings AA must be used, one on either side of the engine, situated on a transverse line which intersects the principal axis.

These mountings will also be called upon to take the greater part of the dead weight and although positioned primarily to give a vertical reaction, the same mountings will be required to furnish a couple to resist oscillations in roll and yaw about the principal axis.

We now have one "fix" on the principal axis. In order to stabilise the system and accommodate oscillations in pitch a further mounting is required which again must lie on the principal axis. If the location of this second fix is suitably selected a percussion system will result (Fig. 3) in which the three necessary elements are: (1) the centre of percussion CP, corresponding to the front mounting; (2) the centre of gravity CG; and (3) the centre of suspension CS, corresponding to the rear mounting. And now when dynamic forces (perpendicular to the axis) are applied at the front mounting they can be completely absorbed there without calling into play any reacting forces at the rear mounting. An excellent example of the percussion system in another field —the cricket field—is provided by the bat itself: The hand grip is at the centre of suspension; the centre of gravity is at CG as before; and the centre of percussion is the point on the blade from which the ball is struck over the fence with such elation for six. No matter how hard the impact at this singular point no reaction is felt on the hands; but a hard stroke from a point appreciably removed from the centre of percussion results in an uncomfortable jarring sensation diminished only by the spring of the handle.

The Centre of Percussion theorem calls for point CS to be situated at a distance L from the point CP as given by the expression

$$L = \frac{Ics}{Md}$$

where I is the moment of inertia about the transverse axis through CS; M is the total mass of the suspended body; and d is the distance between CS and CG. For the present purpose it is more convenient to transpose the expression to the form

$$L = \frac{Icg + a^2}{Ma}$$

using the identities $Ics = Icg + ma^2$ and $d = L - a$.

The additional fix at CS, in conjunction with the mountings at CP now affords couples to resist pitch and yaw respectively about the principal axis. As already mentioned a couple to resist roll is provided by the two spaced mountings at AA so that the torsional stiffness at CS may be made very weak. However all oscillations are generated within the engine about orthogonal axes which are inclined to the principal axis. In consequence both yawing and rolling oscillations, as generated, will have vector components along the principal axis which manifest themselves as roll about that axis, together with vector components normal to it which manifest themselves as yaw. Thus in Fig. 2 if the engine torque oscillations as generated are represented by the vector AB, this will have a roll component BD about the principal axis and a yaw component AD normal to it. Similarly a yaw component generated along AC will have

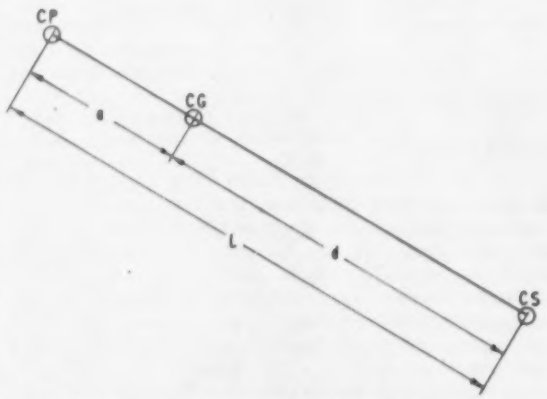


Fig. 3. Diagrammatic representation of a percussion system

a roll component CD and yaw component AD (the scale in this case being different).

The foregoing three point suspension system would precisely accommodate all translational vibrations and torsional oscillations if it were not for one factor. The Centre of Percussion theorem has reference only to force acting normal to the principal axis. But the translational vibrations at the front mounting CP have a vector component in a direction along that axis which must therefore be regarded as extraneous to the system. Nevertheless it can be accommodated at the CP mountings by providing in their design for flexibility in an axial direction along the principal axis.

In Fig. 4 the loads and vibrations to be accommodated by the engine mountings are shown diagrammatically. At idling revolutions the torque oscillations are predominant because their low frequency is then close to the natural frequency of the mounting whilst at higher revolutions the strength of the second order harmonic vibrations overshadow them.

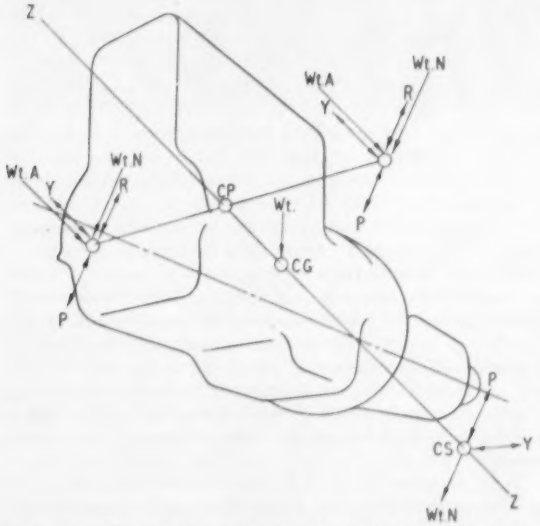


Fig. 4. Diagrammatic representation of the loads and vibrations to be accommodated by the engine mountings

Symbol	Reaction	
R	Rolling couple	
Y	Yawing couple	
P	Pitching couple	
Wt.N	Dead weight	Normal component
Wt.A	Dead weight	Axial component

Little departures from the foregoing requirements can be permitted if the best possible mounting of the four cylinder engine is to be achieved. Such disabilities as are associated with them must be accepted and provided for. Chief amongst these is the location of the front mountings fairly high up at a mid-position in relation to the cylinder block. Unfortunately the front axle cross bearer is usually well forward of this location so that it is not available for directly supporting the mountings and some additional structure is therefore required. And again the steering box often has a prior claim to that location which is not easy to meet with mutual satisfaction to both. As regards the rear mounting, its location is determined by calculation and very little compromise can be countenanced.

The principles of the percussion mounting are equally applicable to the six. The front fix on the principal axis should be positioned as before at midway along the length of the cylinder block. In the absence of a 2nd order harmonic, however, and in view of the lower intensity of the other vibrations in roll and yaw, the mounting can be designed to be much more effective. Indeed considerable latitude from the strict principles can be allowed.

Observations

An historical review of design development suggests that a certain inertia carried over from past practice may be having a constricting effect on the present, when the same conditions do not apply. At the stage when the six cylinder engine had established itself as a pre-eminent power unit the four still retained a wide popularity even as it does today. The demand for greater refinement was not pressing because the four was then ahead of the rest of the vehicle and some improvement in the mounting installation enabled it to keep abreast. Possibly the Lanchester vibration damper, as a unique solution for "converting a four into a six", was for the same reason, ahead of the times, and in any case the patent situation would then have militated against its wide adoption. Modern unitary construction however brings a new element into the problem. At a time when the owner has been educated to expect superlative refinement in comfort and appearance, the problem of boom noise emerges and remains unsolved. And it is the more disconcerting to the manufacturer and distributor alike because it is a will-o-the-wisp. Two successive cars off the line may exhibit a wide difference in boom noise.

Stage thunder in the theatre is produced by shaking a "panel" in haphazard fashion; the noise is correspondingly irregular, as is the intention. If a sustained note were desired no better way could be devised than to impress upon it, whilst suitably supported, a tuned vibration from an outside source and this approaches the conditions in the unitary vehicle.

The problem must be taken in hand at source; the expedient of sound-proofing by the application of flock-felt is inadequate since the large energy input to the panels cannot be absorbed in that manner. The design of engine must be one in which vibrations are either non-existent or else are nullified at the engine mountings. As demonstrated above, the 2nd order harmonic vibrations of the four cannot be completely nullified even with an idealistic percussion mounting possessed of a high value of F/f; therefore the first consideration must be a design of engine that is to the greatest possible extent, devoid of vibration.

Except in the small car where the space under the bonnet is severely restricted, the foregoing would indicate that the six cylinder engine is mandatory. The dissentient will be hard to convert. How shall the Chief Engineer approach his Board with the proposition that the four cylinder engine favoured by them is possessed of a 5th unbalanced piston moving at twice the speed of the four others; how convince them that this complicates his problem of mounting the engine to such a degree that he can give no assurance that the body will not emit boom; that, should it do so, everyone will be in a quandary, that the cost of two additional cylinders more than offsets the risk of jeopardising the market?

For the small car the situation is slightly different. The body panels being smaller have a higher period of vibration and may be presumed therefore to be further removed from the engine period at the lower road speeds. The small engine is a high-revving engine however and this will reduce the margin. By hypothesis there is no room for a six under the bonnet but, nevertheless, the laws of dynamics cannot be made to bow to expediency. The dilemma appears insurmountable. All is not lost however, the means of correction is at hand in the harmonic balancer. It is for the engine designer to incorporate the device with a minimum sacrifice of space and for the manufacturer to reap reward a hundredfold.

The Author understands that Silentbloc Ltd. is developing the percussion system of engine mounting through its designer Mr. J. Nicolaisen. Acknowledgement is made of their investigations.

"Fibre Form" Mouldings

High-strength, Low-density Components Produced from Shaped Pulp-resin Preforms

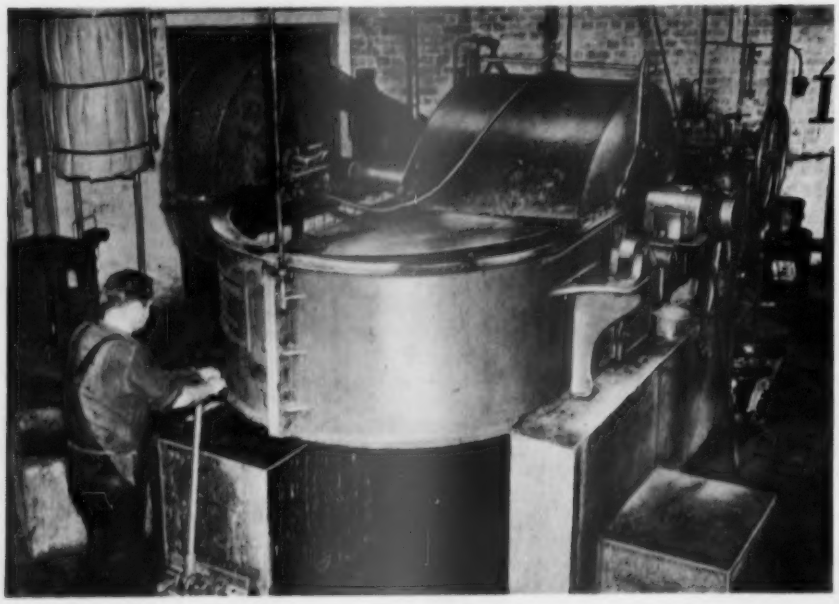
WHILE it may be suggested that a "Fibre Form" moulding, being of a reinforced resin material, belongs to a different category and is not, therefore, strictly comparable with a moulding of powder resin, familiarity with the powder resin product in many different fields of application provokes such comparisons. The reinforced moulding is of higher strength and lower density and consequently has a much more favourable strength:weight ratio. The actual pressing operation is performed at a low pressure, say 200 lb/in², instead of a pressure of approximately 2 ton/in², which is common for powder mouldings. Thus the process offers the possibility of producing mouldings of large area by means of relatively modest press equipment. It is the size of the press platens rather than the power exerted that is the limiting factor. Additional equipment is required, of course, for the preparation of the pulp-resin material and for preforming, but against this may be set the higher cost of powdered resin. The time required for curing is lower, so a higher production rate is possible.

In the production of the preform, papermaking techniques are used. The raw material, Scandinavian wood pulp in the form of sheets of Kraft pulp, is shredded and fed into a beater and defibred in water. This machine consists of a large bath-shaped tank, lined with tiles, in which a heavy corrugated roller extending over half the width of the tank is rotated over adjustable stationary blades. Belt-driven by an electric motor, the roller draws a thin stream of pulp between it and the blades and recirculates it in the tank. The synthetic resin, usually a phenolic resin in either a finely divided state or as a solution, is added and also any pigments or dyes.

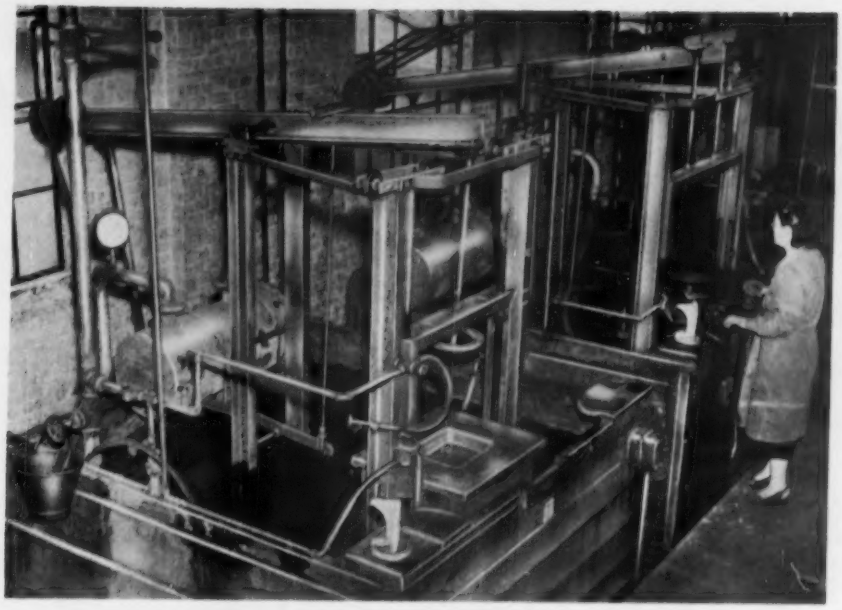
The "degree of beating" is carefully controlled to suit the required characteristics of the end product. Too much beating can impair the felting properties of the pulp, and lower the "freeness" of the pulp. The lower the freeness, the greater is the tenacity with which the cellulose fibres hold water. It is also necessary to control the acidity of the resin-pulp mix. Too alkaline a pulp will adhere to the felting tool ; too acid a pulp will adversely affect the moulding process. Freeness is ascertained by a test in which a specific volume of pulp is timed on its passage through a screen similar to that to be used in the felting process. After beating, the resin-pulp mix can be stored in tanks or drums for a period of from one to two weeks. From these common stocks blends of pulps are prepared for specific batches of work. The blends are thoroughly stirred on a battery of mixing machines to ensure consistency in processing and uniformity of the end product.

" The "shape" or felt", as the preform is termed, is produced in a felting vat by a vacuum process. These vats each have a capacity of about 500 gallons and the required quantity of resin-pulp mix is added to water to give a slurry having a solids content of from 0·5 to 2·0 per cent. The fibres are maintained in a uniform suspension by constant agitation of the slurry by means of compressed air. A felting tool, comprising a perforated metal support covered with a wire screen approximating to the interior shape of the desired preform, is mounted in a mould box reciprocable in vertical slideways. By means of an overhead beam mechanism it is submerged at predetermined regular intervals into the slurry. The tool is connected to a vacuum line and when submerged suction is applied and the random fibres are drawn to and deposited on the exterior of the felting screen.

Time required for felting varies from 5 seconds to 30 seconds and the thickness of the deposit varies approximately



The beater in which the Kraft pulp is defibred and the synthetic resin added to form the resin-pulp mix



The battery of 500-gal felting vats in which the diluted pulp slurry is maintained in a uniform suspension

in proportion to the period of immersion. When the tool is raised from the slurry the shape still contains a fairly high proportion of water and is deformable but is, nevertheless, of sufficient strength to withstand handling in the shop. On removal from the tool the shape is immediately weighed on a spring balance, shown standing on the side of the vat in the illustration. A tolerance of from 0·3 to 1·0 per cent of the total weight is commonly allowed.

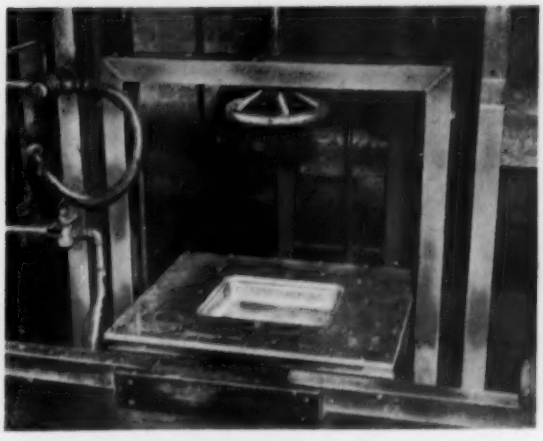
It is then necessary to dry the shape. This is usually carried out in tunnel ovens and on emergence the shape may have a resin content of from 15 to 40 per cent. In this condition it is quite stable and can be stacked and stored until required for final moulding. The drying time may range from 2 to 4 hours.

As it is taken from the felting vat, and before drying, the shape may be subjected to a cold pressing operation. Termed "presqueezing", its purpose is the densification of the felt and the partial removal of water. An alternative method, known as "die drying", is to press the shape in heated dies. In this way the shape is densified and dried simultaneously.

It will be appreciated that in the drying operation the curing of the resin will be commenced. Actually, after drying the resin has been advanced approximately to the condition of the commercial moulding powder and is ready for final moulding in heated steel dies in conventional hydraulic presses. The technique does not differ from that employed for powder moulding except that, as moisture is released during the operation, several lifts may be made to permit blowing-off of surplus moisture. Curing cycles vary considerably, depending on the resin content. For preforms having a relatively low resin content, the final cure time may be a matter of seconds only.

Superficially regarded, the preparing and preforming processes would appear unduly complicated and likely to be uneconomical in comparison with moulding powders. Such views are corrected, however, if consideration is given to finished articles produced by the respective methods. The

A complete vacuum felting tool in its submersible mounting on a felting vat



Demister nozzle formed by two mouldings stapled together at the flanges

This tote tray for the textile industry measures 27in × 18in

Single-curved and double-curved panels covered with leathercloth

advantage conferred by the high strength:weight ratio of the preformed article is readily apparent. It has been stated that the factory cost of a preform is below the price of the moulding powder required to produce an article of equivalent strength. This price differential increases steeply in favour of preforms as the size of the article increases.

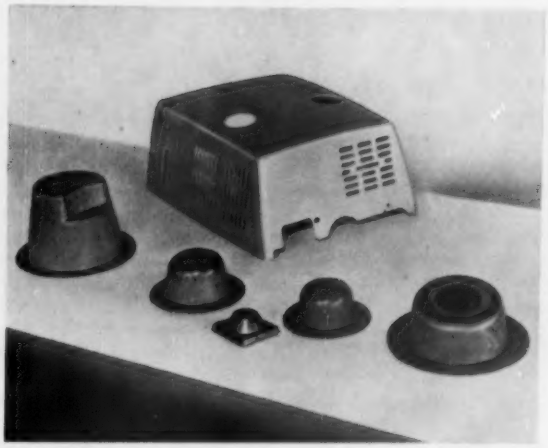
The curing cycle for the preform moulding is faster and is carried out at a much lower unit pressure. This is possible as the resin composition is not required to flow. The moulding of a preform merely densifies the shape by compression and bonds the interlocked fibrous structure. As compared to powder mouldings Fibre Form mouldings, are competitive while if compared with other types of reinforced plastics they offer advantages in respect of low cost and a high production rate.

A limiting factor of the vacuum-felting process is the thickness of the felt that can be produced. The knitting of the fibres on the forming tool becomes progressively less as the felt builds up and the suction is thereby lessened. Eventually the effect of the vacuum is reduced to zero and no more fibres are deposited on the tool. When thicker felts are required for articles of heavier section resort is had to the pressure-felting method. A specific volume of pulp slurry of a relatively heavier consistency is confined in a felting tool formed to the outer shape of the preform. Pressure is applied by a piston unit and the water is forced out through the wire screen leaving the felted fibres inside the tool. Felts produced in this manner, however, are not so uniform in structure and, as the fibres are not so well interlocked, are less strong.

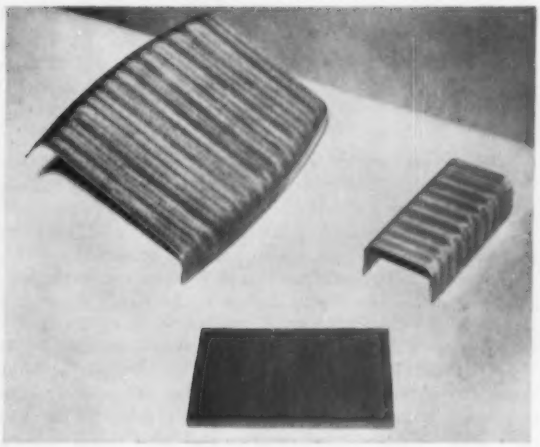
There is also a limit to the amount of resin that can be incorporated by the beater dispersion method described. When a higher resin content is specified, the resin can be impregnated into a pulp preform after it leaves the drying oven. The impregnated preform must then be passed through a second oven, to remove the resin solvents and to advance the cure of the resin, before being finally moulded

The only restriction on the size of preformed mouldings is that imposed by the capacity of the production equipment. At present, the largest presses at the London works of the manufacturers, Fibre Form Ltd., Garratt Mills, Trewint Street, Earlsfield, London, S.W.18, are Bradley and Turton 175-ton machines having platens 30in × 30in. A special tote tray for the textile industry is illustrated as an example

Some of the hydraulic moulding presses on which the preformed shapes are finally cured. Stacks of preforms will be seen on the right



Various moulded caps and covers and an apertured casing



Curved and flat moulded panels finished with natural wood veneers

of a large moulding having a fairly deep draw. Its overall dimensions are 27in long and 18in wide. The larger the moulding, the lower is the unit cost as compared with that for alternative materials such as powder or laminated plastics, or timber. Long production runs and multi-impression tools, frequently possible in view of the low moulding pressure required, will further reduce costs. Presses with 36in × 36in platens are installed in the firm's Wolverhampton works.

After the final cure the article is dimensionally stable and can be trimmed, drilled or punched without tearing or shredding of the edges. This facility for simple post-moulding operations obviates any need to complicate tooling or to lower production rates in order to mould in apertures or slots. Moulded parts may be assembled by stapling at the flanges, as in the case of the demister nozzle illustrated.

Since in the moulding process the fibre structure is not ruptured or caused to flow, the resulting product is uniformly strong and has a good surface free from irregularities. The strength is comparable to that of glass-reinforced plastics. Adequate strength is obtained in thinner sections than are practical in powder mouldings, with a consequent saving in weight and cost. It is commonly moulded to a thickness of from 0·040in to 0·070in. The density of a Fibre Form moulding is approximately one-seventh of that of steel. In substituting a moulding for a steel pressing the section would probably be twice that of the original steel article but, even so, it would be only one-third of the weight.

In the past, a drawback of the fibre-resin moulding was that the surface, although smooth and not unpleasing, was of a uniform grey colour and not sufficiently decorative for many applications for which the material, by virtue of its properties, was eminently suitable. A wide range of finishes is now available as a result of continued development by the firm to eradicate this handicap. During the moulding process, or as a subsequent operation, a thin decorative sheet can be moulded on the material. These sheets include natural wood veneers and a variety of printed papers simulating woods, leathers, fabrics, and other materials. They are heavily impregnated with resin which during the moulding operation, under the influence of heat and pressure, imparts a permanent glossy finish to the article.

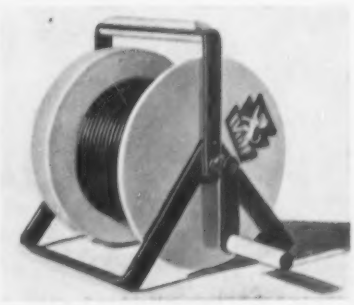
An important advantage of veneered or paper-surfaced pulp-resin mouldings is that they can be produced with curved or double-curved surfaces and in a variety of simple shapes, whereas the decorative laminates are only available as flat sheets. Thus, the designer is offered the possibility of improving appearance and at the same time reducing both production cost and time.

Another development is the coating of the moulding with fabrics, such as impregnated printed cottons, linens or leathercloths, and also with unsupported PVC sheeting. Both the article and the covering material are coated with a resin adhesive and, by a specially adapted moulding technique, the material is stretched over the article to give a permanent, wrinkle-free finish.

Mouldings can be sprayed with paints or lacquers. An attractive finish is also obtained by metal spraying.

For motor vehicles many suitable applications of fibre-resin mouldings occur in the interior trim, such as door and roof panels, instrument and facia boards, glove boxes and door pockets—which can be coated with flock, window frames and steering column shrouds. Other items, not requiring a decorative finish, are heating and ventilating conduits, battery support trays, demister nozzles and various covers, protectors and component housings.

Portable Power Point



OF particular use on maintenance work, this new Wolf unit enables two electric tools, or a tool and a light, to be operated from a single remote supply point. It comprises a drum rotatably mounted on a stand having a carrying handle; the capacity of the drum being 100 yd of 23/·0076, 90 yd of 40/·0076 or 60 yd of 70/·0076, 3-core cable. The cable end is connected to a terminal block in the drum hub which, in turn is connected to a pair of 13 amp, 3-pin, shuttered, flush-type sockets in the drum cheek remote from the winding handle. Net weight is 9¼ lb.

COOLANT AND LUBRICANT APPLICATION

Aerosol Sprays for Metal Cutting Operations

FOR the majority of machining operations it is necessary to use either an oil or a coolant solution to lubricate the cutting edge of the tool and to dissipate the heat generated in the removal of stock from the workpiece. In all cases the problem is to ensure that the coolant is actually applied to the tool edge and in sufficient quantity to prevent overheating. Flooding of the work with coolant is commonly resorted to but even this cannot be relied upon to achieve the desired result, particularly if the machining speed is relatively high.

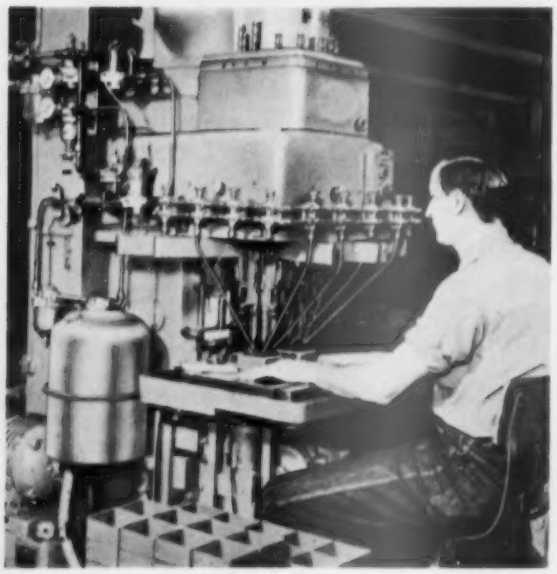
In any case flooding is most likely to be wasteful as the coolant is flung off the work and, unless adequate shielding is provided, is deposited on the machine and the adjacent floor. Coolant is carried away on the chips and if recovery is desired a special spinning operation is necessary. The work must be allowed to drain and in some instances must be washed before a subsequent operation is undertaken.

These disadvantages are eliminated and substantial economies are effected by the use of one of the two compressed air operated coolant systems introduced by C. A. Norgren Ltd., Shipston-on-Stour, Warwickshire. With these systems designated "Spray-Lube" and "Mistcool", the cutting oil or coolant is atomized, entrained in a flow of air obtained from the shop air line, and delivered by way of either a rigid or a flexible conduit to the cutting area. The quantity of coolant, the volume of air, the character of the spray, and the air pressure can each be regulated to suit the delivery to the requirements for specific operations.

Spray-Lube system

The essential feature of the system is that by atomizing the coolant a greater fluid surface area is exposed and thus heat can be absorbed more rapidly than by a continuous stream of coolant. The compressed air used to convey the coolant provides an additional cooling factor. Each spray nozzle is located closely adjacent to the tool and suitably directed to reach the critical cutting area. Commonly they deliver from the underside of the tool in order to penetrate the interfaces of the tool and work below the shearing point of the chip. In multiple tool layouts individual metering valves can be provided for each nozzle. Since the coolant is finely divided it decomposes almost instantaneously on the hot chip to form a dry lubricant. As a consequence, flying chips do not stick to the machine, chip collection and disposal is facilitated and machine cleaning is simplified.

Spray-Lube equipment with twelve individually regulated feed nozzles on multiple tapping machine



Basic equipment for the Spray-Lube system comprises a filter for the compressed air supply from the shop line, a pressure regulator for the air line to the mixing valve, and a pressure regulator for the air supply to the coolant reservoir. This second supply gives a pressurized feed of coolant from the reservoir through a liquid filter to the mixing valve. When cutting compounds that have a tendency to separate are used, a simple agitating device is employed. The pressure air is fed to the bottom of the reservoir through a tube attached to the head fitting and air is vented from the top of the reservoir at a rate adjusted to provide constant agitation of the contents. When agitation is not required the air vent is reversed in the head fitting. The reservoir is equipped with a contents level glass and a ball check valve prevents fluid feed-back when the system is de-pressurized.

Air line and fluid line are connected by copper tubes to the mixing valve to which the spray tube and nozzle is attached. This unit has a screwed needle valve to regulate the rate of coolant feed. In multi-point applications requiring individual control of the spray at each point, further mixing valves are connected in parallel. Should a single control be adequate, a 4-point or 8-point aerosol distributor can be fitted between the mixing valve and the individual spray tubes. For economy in operation it is usual to include solenoid valves in the air and fluid lines. These are energized by means of switches actuated by machine movement and suitably timed so that coolant is delivered only during the actual cutting operation. The air control can, of course, be arranged to clear away swarf from the work before, after, or both before and after the cutting operation.

A typical Spray-Lube installation for a multiple tapping machine is illustrated. This machine handles a wide variety of work, some items of which require the simultaneous operation of twelve taps. A battery of twelve mixing valves mounted around the tapping head is fed by a $\frac{1}{2}$in diameter copper tube from the fluid filter and from each mixing valve a 16in length of $\frac{1}{4}$in diameter tube with a spray nozzle is provided to direct the lubricant to the individual taps. Solenoid valves in the air and lubricant lines bring the spray system into operation only when the taps are approaching and actually cutting. Five gallons of cutting oil are held in the reservoir.

On the set-up shown three holes are tapped in each of two switchbox covers. Actual production rate on the machine was increased by 30 per cent over that obtained when ordinary flood lubrication was provided. With that system a second operator was required to stack the components for draining and to wipe off lubricant before they could be progressed for subsequent operations. Working conditions were much improved and the saving in lubricant consumption was substantial. A standard Oil-Fog lubricator, shown mounted on the side of the machine, is also used to lubricate the transmission gears in the tapping head.

Vertical turret lathe supplied with coolant from a centralized Spray-Lube distribution station for roughing and finishing cuts on aluminium tyre capping moulds

In another example the Spray-Lube equipment and control panel is centrally located to serve several machines. The vertical turret lathe shown is employed to machine the ribbed insides of cast aluminium tyre-capping moulds about 90in in circumference. Roughing and finishing cuts are taken by two one-piece, 14·25in wide, face-form tools. Initially, the work was machined dry and later compressed air was used as a coolant. Surface cutting speed was 45 ft/min and the feed was manually controlled at the discretion of the operator. By either method there was persistent welding of chips to the tool edge and the surface of the work was galled.

With the Spray-Lube system air at 60 lb/in² and Dypral coolant, in a ratio of 1:15 of water, at 62 lb/in² is mixed and sprayed from four nozzles over the top of the tool cutting edge. The output of the machine on this work has been almost doubled. Surface finish has been improved, chip-welding and galling has been eliminated, and tool maintenance has been reduced. The surface cutting speed has been raised from the previous 45 ft/min to 75 ft/min and feed is now constant at 0·006in per revolution.

Two ¼in diameter copper supply lines bring air and coolant 30 ft to the mixing valve mounted on the side of the machine. From the mixing valve 14 ft of flexible hose fitted with a quick-connection coupling conduct the mixture to one of the small distributing manifolds attached to the tool posts. Each of these manifolds carries four $\frac{1}{16}$in diameter spray lines, about 12in long and positioned to give full spray coverage over the cutting edge of the tool. Chips are removed from the table by a large diameter suction hose.

Mistcool system

For the single-point application of cutting fluids somewhat simpler equipment, giving the maximum economy in initial and running costs, is available. Termed the Mistcool system, a unit consists of an air filter, a pressure regulator, and an oil-fog lubricator, all of which are standardized units. A complete installation will include additionally a solenoid valve for spray control by the machine, the necessary ¼in diameter copper pipe or flexible hose for connection, and an adaptor with a $\frac{1}{16}$in diameter bore brass tube to direct the spray from under the cutting edge of the tool. The system is particularly suitable for single-point lathe tools, for grinding carbide tools, and for machining plastics.

Normal capacity of the oil-fog lubricator is one-third of a pint but a 1¼ gallon reservoir can be supplied. Various models of lubricator are available, suitable for soluble oils, for neat cutting oils and for oils containing graphite, molybdenum sulphide, or similar admixtures. Normally the oil is fed through a sight drip feed to the air line venturi and directly entrained, but for neat cutting oils a special micro-fog lubricator is additionally available. In this unit the oil mist is blown into the air space above the oil level in the bowl and all but the finer particles of oil separate out and are returned to the reservoir. The fine particles, 2 microns or less in diameter, are drawn off by the main air stream. Only about 5 per cent of the oil passing the sight feed is actually delivered at the spray nozzle.

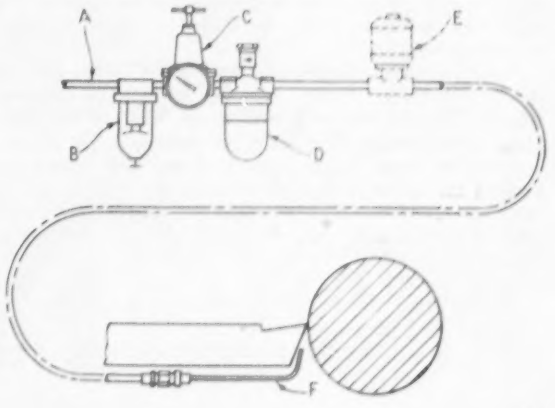
For a wide variety of machining operations substantial economies of cutting oils, increased tool life, reduced production times and improved work finish can be attained, together with attendant supplementary advantages of easier working conditions, cleaner machines, and dry swarf removal and recovery. Following are a few examples of recent applications of the Mistcool system.

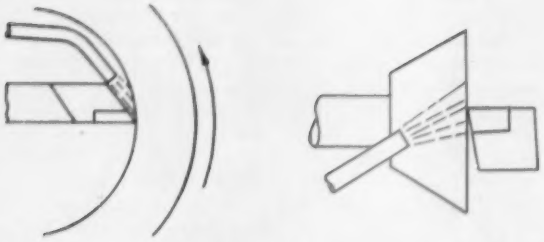
A component in 35—40 ton steel, with a bore 95 mm diameter was internally screw-threaded 1·5 mm pitch for a distance of 1·0in on a Cridan screw-cutting lathe. A standard carbide-tipped tool was used, the work speed being 900 r.p.m. With Mistcool equipment feeding Vaughan's No. 15 broaching oil at the rate of one drip per second, tool life was increased by 400 per cent, the number of finished components obtained between tool settings being raised from the previous 20 to figures of from 70 to 99. There was no objectionable smell of heated tallow while working and at the end of one day's production output was three hours ahead of schedule as a result of the saving in re-setting time.

Tungsten carbide blades for an 8in diameter face milling cutter were ground with a Bakelite-bonded diamond cup wheel 5in diameter. The operation was done dry and 0·015in stock was removed in 30 passes making cuts of 0·0005in. With one Mistcool spray of 60 per cent paraffin and 40 per cent Delapena honing oil at the rate of two drips per three seconds the same amount of stock was removed

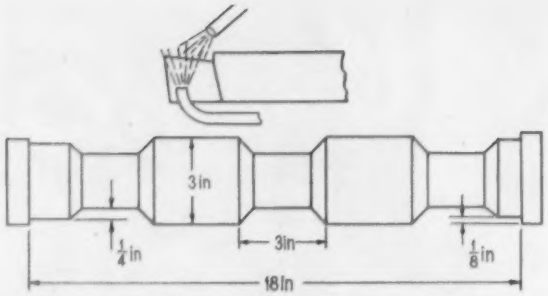
Diagram of Mistcool equipment for turning operation. The solenoid valve is an optional extra

A—air supply, B—air filter, C—pressure regulator, D—oil-fog lubricator, E—solenoid valve, F—spray tube





Left, internal threading on a Cridan screw-cutting lathe. Right, grinding tungsten carbide milling cutter blades with a diamond cup wheel



Turning on a Redman-Churchill semi-automatic hydraulic copying lathe, using two Mistcool sprays

in ten passes making cuts of 0·0015in. Wheel speed was not changed. Although a substantial saving in time was effected by the reduction of the number of passes from 30 to 10, there was no wheel tear, or burning or thermal cracking of the blades.

Shafts of EN-361 50-60 ton steel were turned on a Churchill-Redman semi-automatic hydraulic copying lathe at a rate of 35–40 per regrind. The carbide-tipped tool was flood cooled, feed was 9in/min and speed was 475 r.p.m. With two Mistcool sprays delivering one or two drips per second of Dicks soluble oil in a 1:20 mix, and using the same feed and speed, output was raised to 70 per regrind.

As regards the machining of plastics, the nature of those materials gives rise in rapid abrasion of the tools and necessitates regular stopping of the machine for tool changing. Mistcool cooling has greatly extended tool life between regrinds on this work. An illustration of equipment mounted on a high-speed router machining plastics is given on the Contents page.

Aerosol spray systems are also used for the timed application of die lubricant to either the blank or the tools in pressing and forming operations.

Precombustion-Chamber Diesel Engines

German Investigations into Performance

COMBUSTION in precombustion-chamber diesel engines is discussed by H. Hoffmann in M.T.Z., May 1956.

Characteristic curves of two Diamler-Benz engines illustrate the outstanding performance of precombustion-chamber diesel engines. Such engines can burn a wide range of fuels without modification to the combustion chamber. These considerations lead to an examination of precombustion-chamber design and current types of precombustion chamber; experimental results with variants of the precombustion chamber and the process of combustion in precombustion-chamber engine are reviewed.

Diagrams show a pre-war Diamler-Benz commercial engine of the OM 67/4 type and injection-nozzle modifications which led to improved performance. Similarly, stages in the development of a modern production-engine, OM 312 are shown, the latest version of which employs a turbulence outlet nozzle. These precombustion-chamber engines use single-hole injection, while the OM 315 engine employs multi-hole injection.

Engine-running characteristics are affected by the ratio of the pre-chamber volume to total combustion-chamber volume. Results achieved with direct-injection and precombustion-chamber versions of the same OM 312 engine, compression ratio being kept constant, are compared. Despite the increased fuel consumption involved by use of the precombustion-chamber principle as such and by a high ratio of pre-chamber volume to total combustion-chamber volume, ratios of up to 40 per cent are favoured, chiefly in order to prevent Diesel knock. In the OM 312, violent knocking occurs with a small pre-chamber, and injection for optimum performance has to start at 27–28 deg B.T.D.C. With a pre-chamber 66 per cent larger, the engine run quietly; for optimum performance, injection begins at 24 deg B.T.D.C. At full load, the wall temperature of the small pre-chamber is 310 deg C. and that of the large one 560 deg C. Reduction of the quantity of fuel injected to a quarter of the normal full-load quantity lowers the wall temperature by only 74 deg but lowers the exhaust temperature by 500 deg. Artificial raising of the wall temperature of the small pre-chamber to 540 deg C. (close to thet of the large one) has practically no favourable effect on knocking. Considerations of the significance of these findings together with observations on multi-hole-injection engines indicates that the high outlet energy from the pre-chamber necessary to prevent knock is obtainable by a high ratio of pre-chamber colume to total combustion space and a restricted cross-section of the outlet. Combinations of both conditions are possible though not always successful.

Idling knock has causes quite different from those of the running knock so far referred to and must be countered by different methods. This fact is illustrated from the example of a private-car engine, OM 636. Some experiments are also reported on the influence of the spreading-out of the fuel spray on quiet engine running.

A study of the combustion process in the precombustion-chamber engine compares the influence of the turbulence injection nozzle in the OM 312 engine with that of the smooth nozzle in the earlier version of the small engine. It is found, *inter alia*, that: (1) the shorter the ignition lag, the smaller the tendency to knock; (2) knocking originates in the main combustion chamber and not in the pre-chamber; (3) the more fuel available for combustion before ignition starts in the main chamber, the greater the tendency there is for knock to occur.

Hydraulic Transmissions

PART I: *A Survey of the Basic Principles Involved*

R. Dean-Averns

THERE is an increasing tendency to introduce into the transmission some form of cushioning medium, not only as a means of smoothing out the transmission of power, but also to reduce the possibility of torsional vibration throughout the system, and to produce a softness similar to that in evidence in hydraulic transmissions which occurs during gear change at the instant the driver releases the accelerator pedal. In this respect the hydraulic coupling or fluid flywheel and the hydraulic torque converter were considered as possible sources from which research could commence and it is not, therefore, surprising to find this tendency at present assuming considerable proportions. In order more fully to comprehend the functions of these modern applications, it is necessary to recapitulate very briefly the working principles of both systems, since the hydraulic coupling and the torque converter must not be confused.

Hydraulic coupling

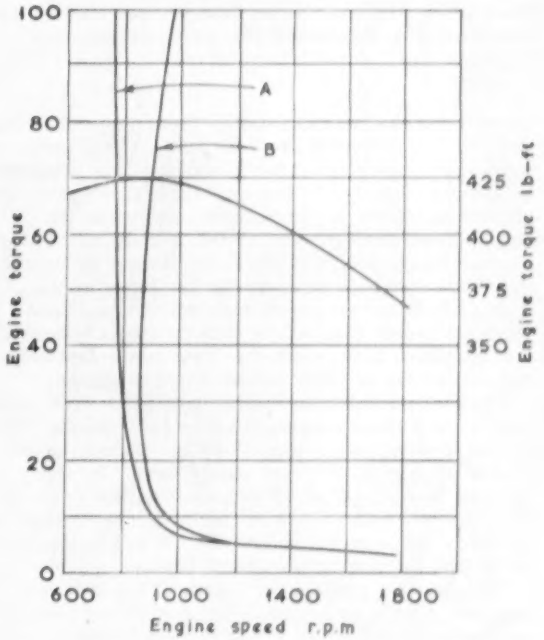
The hydraulic coupling comprises essentially two main members, the impeller and the runner or driven member, which have straight radial vanes forming a series of cells in the castings. The vanes are not equal in number on each rotor although the difference of two or even one is sufficient to prevent all the vanes in one member passing those of the opposite rotor simultaneously. Normally the two rotors are attached, one to the engine crankshaft and the other to the gearbox input shaft, and when in position together, they form what is termed a *torus*. In designs which include a core guide ring, the ring is located nearer to the outside periphery of the torus than to the inner diameter. The reasons prompting this feature are many, the main one being to ensure that the area of the path of fluid flow is as nearly as possible uniform around the circuit with frictional and eddy losses as low as possible, in other words, to ensure the highest efficiency. The volumetric capacity of the torus is not completely occupied by fluid, although it must be appreciated that the percentage volume employed has a direct influence upon the performance of the coupling, see Fig. 1.

In these brief notes, it is not possible to describe to the full the individuality expressed in the various approaches to this somewhat complicated development; particularly the hydraulic advances. Reference can only be in general form, embracing common features.

The performance of an internal combustion engine is limited, but the demands made upon the vehicle vary considerably from climbability to speed, whilst the horse power, torque and efficiency curves of the engine are maintained over a comparatively small range of engine speeds. Whilst it is the aim of the designer to maintain engine speed as constant as possible over as wide a range and to match this range with the most economical operating vehicle speed, some compromise is necessary for the maintenance of the desired torque to compensate for the variation in tractive effort and resistance occasioned by the varied duties the vehicle is called upon to perform. Such adjustment is usually effected by employing an intermediate transmission as a means of varying the velocity ratio, whilst transmitting sufficient power.

This necessity has been the source of many ingenious developments from the countershaft and belt drive of the old cycle car to the more recent conventional gearbox and later to the employment of fluid transmission. It is with this last development that these notes are mainly concerned. There have been many attempts to devise a form of transmission which does not entail toothed gearing, with varying degrees of success. Such transmissions fall within one of three categories: electric, hydraulic and mechanical.

The electric type comprises generally a generator, supplying current to a motor that is in turn coupled direct to the final drive or axles. This system did not, however, achieve great popularity at least in road vehicles although it is not unheard of today in some railcar units of modern design.



Curve A—full throttle stalling speed 750 r.p.m., 30 per cent fluid filling
Curve B—full throttle stalling speed 950 r.p.m., 50 per cent fluid filling

Fig. 1. Typical curves showing the effect of volume of fluid and slip at full engine torque.

Electrical controls have been applied to epicyclic boxes, such as the Cotal gearbox, for instance, which had its various epicyclic trains braked by electro-magnetic control to obtain the desired ratio. Epicyclic gearing provides a simple and economic form of transmission and is not without its adherents today. Such a gearbox usually comprises essentially a sun wheel, planet and annulus. The velocity

or gear ratio is dependent upon the state of rest or motion of a third member. There are proprietary designs of epicyclic gearbox which render the necessity for a separate clutch less vital. In such designs the ultimate fixed member is free, and is brought to rest when desired, the driven member being gradually speeded up, the clutch action being obtained by the gearing itself.

The main function of the gears in the orthodox gearbox is to transmit motion from input shafts with uniform velocity and with as little shock loading as possible. If the moving parts or those to be engaged be synchronized in speed the latter condition is considerably assisted, and designs of synchromesh mechanisms have been produced in efforts to accomplish this feature. Such designs may generally be classified in two categories: (*a*) the inertia lock type, and (*b*) the constant load type of synchromesh gear. In both types the object is to provide a condition whereupon positive engagement of the gears is prevented until such times as both gears are rotating at similar speeds. Cone clutches are employed in both systems to date, the main difference being that in (*a*) it is not possible to *crash* engage the gears by too much pressure on the control mechanism, whereas in (*b*) it is.

In the inertia lock synchromesh gear, any endwise movement of the synchronizing cone clutch brings into engagement the inertia lock, until there is no relative slip between the two cones. The time taken to synchronize is inversely proportional to the load applied, since the larger the load on the controls, the greater is the force between the two cone clutches and in consequence the greater the lock load.

$$\text{Torque} = \frac{k\pi n}{6t}$$

where k = radius of gyration, n = revolutions per second, and t = time in seconds to synchronize. The difference in speeds of the two cones at the time of change is occasioned because both cone and drum rotate slightly until a projection on the drum makes contact with the mainshaft spline. The engaging dog exerts pressure on the drum by virtue of the chamfered edge when it is impossible to move the engaging dog further until the drum torque has decreased, which it does gradually as the speeds approach synchronization, to the point whereat it is just less than the torque between the two chamfered faces, when the drum moves forward and permits the dog to follow through and thus engage.

The constant load synchromesh type, as has been stated, does to some extent permit gears to be crash engaged before synchronization takes place, if either machine or heavy load is applied to the gear change lever. In this type, pressure from the change speed lever is applied at the outer ring. This is transferred to the inside member through the groove in the outer ring and via the ball and spring, which holds both members in the neutral position.

Through this medium the speeds of the gear and the engaging dog are synchronized, but if additional pressure is applied to the outer ring, the springs are further depressed and allow the ring to slide over the inner member and engage the gear. It should be pointed out that the angle of the cones and clutch are of considerable importance as upon these angles depends the loads required for synchronizing. These angles usually vary between 6 deg and 7½ deg, which also allows sufficient longitudinal movement to maintain the clearance between the cone face and drum, usually 0·005in. It will immediately be appreciated how vital correct lubrication will be, including the design of grooves in the cone, as these can either throw off the lubricant or hold too much and cause slip. These angles control the speed at which synchronization takes place, while the clutch performs the function of changing the inertia of the moving parts from their existing running speeds to that of the new gear velocity. It may not be out of place to set down the fundamental formulae required in preliminary investigation into such a design.

The torque required for synchronization is usually force = mass x acceleration:

$$\text{Torque} = \frac{M}{g} \times k^2 \times \frac{2\pi n}{12t}$$

where k = radius of gyration, in
M = weight, lb
g = gravity 32·2 ft/sec²
n = rev/sec
t = synchronization time, sec

The torque required to synchronize the various speeds and which is applied by the cone clutch is:

$$T = \frac{I\pi n}{6t}$$

or, for all parts:

$$T = \frac{\pi}{6t}(In + I_1n_1 + I_2n_2, \text{ etc.})$$

and the spring load $F_s = \dfrac{F \tan \theta}{N}$

where I = radius of gyration squared multiplied by the mass, N = number of balls and $F = T \sin \theta/N$.

Epicyclic gearing

Perhaps the best known gearbox of this type, always excepting the evergreen Ford type, is the Wilson preselective box in which first gear is obtained through a simple epicyclic train, the annulus being held with the drive from sun wheel to spider-mounted planets. Higher ratios are obtained by releasing and driving the annulus on the first speed train, which is speeded up again for each successive ratio. All the drives and ratios are selected by the application of a band brake to its appropriate annulus. After a period in which it was thought that the four-speed automatic epicyclic gearbox was to be the ultimate connection between engine and axle, the fluid flywheel and torque converter became highly favoured in the United States of America, and it is now offered in various forms on most American cars, sometimes as an alternative to the more usual type of gear-change.

However, in considering the potentialities of this transmission, due regard must be made to British road conditions. Fuel in the United States is cheaper than it is in this country and a slightly increased consumption is not therefore a serious handicap to a vehicle. The question of engine power, and therefore power:weight ratio enters into the consideration. It is known that fuel consumption of a car with torque converter transmission varies more with operating conditions than does that of a car with ordinary clutch and gearbox. On the open road with but few stops or changes, consumption increases but little, whereas it falls off rapidly when driving in traffic and on winding roads such as predominate in England.

The governing principles may be stated as in the following pages. Most American cars that are fitted with a converter transmission have a minimum b.h.p. of 100, for if the power:weight ratio is low, the torque converter does not show up at all well. In fact, a fluid flywheel with an ordinary three-speed gearbox would be just as acceptable for performance comparison. This would not be compatible with American ideas.

Not so long ago, the employment of hydraulics was mainly confined to the transmission of power or force by pressure and for remote control of mechanisms. These were the hydrostatic systems as exemplified in the power hammer.

Nowadays, hydraulic systems form two general classes; they are the hydrostatic and the hydro-kinetic systems. The former is characterized by pressure differences and

flexibility of design, while the hydro-kinetic system derives its results by virtue of changes in velocity of working medium, or fluid. The system consists of a pump and an oil turbine, and power is transferred from the pump or driving shaft through the circulation of fluid between the impeller and runner, or turbine. Such transmitters depend for their class upon whether or not they include intervening static members between the impellor and runner. If they do not, they can obviously only transmit impeller torque and are therefore referred to as hydraulic couplings. However, if static members are included in the working circuit, then the torque of the impeller can be multiplied and the system is classified as a torque converter.

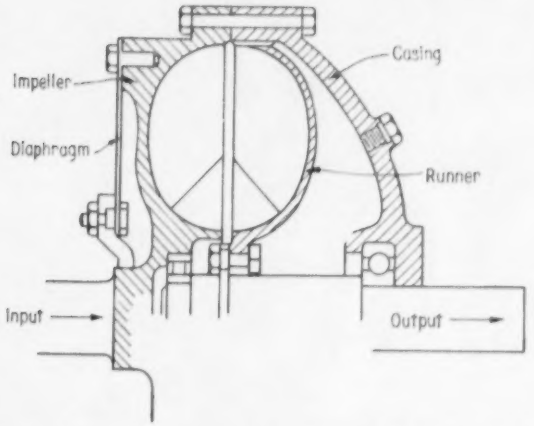


Fig. 2. Diagrammatic sketch of the components in a hydraulic coupling

This may be further explained by stating that as the sum of the torques acting on any system must be equal to zero, then in order to obtain any difference between input and output values, the converter requires some other member interposed to divert the fluid and to provide reaction in so doing.

As interposed in an automobile transmission, the fluid drive is particularly suitable to deal with the various characteristics involved, more especially in the case of a diesel-powered vehicle, although not confined to this type of power unit. Some of the qualities it possesses are:

1. It permits a smooth take-off on starting from rest under heavy load.
2. It provides freedom from *snatch*, thus relieving transmission of shock loads.
3. It increases the flexibility of control of the vehicle.
4. In the event of the vehicle stalling, a coupling limits the torque to a safe figure, which prevents stalling of the motor and damage due to the flywheel effect.

The relation between power, torque and revolutions is stated in the expression:

$N = C_1 n^3$

where N = power, n = rotational speed and C_1 = a constant, from which it will be seen that the diameter of the coupling influences the output to a considerable degree. The expression $N = Cn^3D^5$ illustrates this extent, showing that the power transmitted increases as the fifth power of the diameter, in inches. Different characteristics are obtained by variation in blade design for the impeller. The conversion of torque is determined by the blading of the impeller and turbine wheels.

The simplest form of transmitter is the hydraulic coupling, a typical example of which is shown in Fig. 2. In this, the impeller is attached to the input shaft, while the casing, which forms a reservoir for fluid when the coupling is stationary, is bolted in turn to the impeller. The runner is attached to the output shaft, and motion is imparted to it in the following manner.

The path of the fluid is, of course, rotational but upon first movement of the impeller it is subjected to the effects of centrifugal force and therefore flows outwards in a radial direction leaving the impeller and entering the driven member. By this flow it displaces fluid from the inner circumference of the small cells, formed by the vanes, into the impeller. So long as there remains a difference in velocity between the driving and driven members, this circulation continues inside the cells, due to there being a greater centrifugal force acting upon the fluid in the higher velocity impeller than there is in the runner or driven member. The fluid travels with both the impeller and runner round the axis of the coupling, at the same time flowing round the core or central portion of the torus in a vortex motion, Fig. 3. There are promoted, therefore, two motions of the fluid, resulting in the development of a certain amount of kinetic energy.

It has previously been mentioned that the central core guide ring is disposed a little nearer to the outer periphery of the torus than to the inside diameter, in order that the area of the flow path of the fluid may be constant. In this case, there is no change in the velocity of the vortex motion, but the velocity of the oil round the axis of the coupling increases as it flows outwards from the inlet to the outlet of the impellor cells, again decreasing during its passage inwards in the cells of the driven member. As the velocity

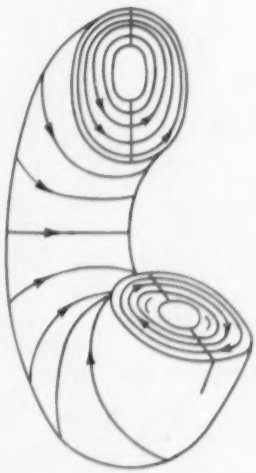


Fig. 3. Flow of oil in the torus

in the cells of the impeller increases a resistance to impeller motion is produced and therefore the kinetic energy from the impeller must increase. This energy is transferred to the driven member, as the fluid velocity decreases in that rotor, and causes the oil to impinge in a forward direction against the vanes to produce the required power at the driven shaft. During the period of reversal of torque, such as may occur when the accelerator pedal is released for gear change, there is a momentary change in direction of the vortex motion of the fluid, since at that period the car is driving the engine, or, in other words, the runner drives the impeller. It is at this nodal period that advantage should be taken to change gear, as the fluid will revert to its former direction the instant the accelerator pedal is again depressed.

In an automobile application, it is essential that the

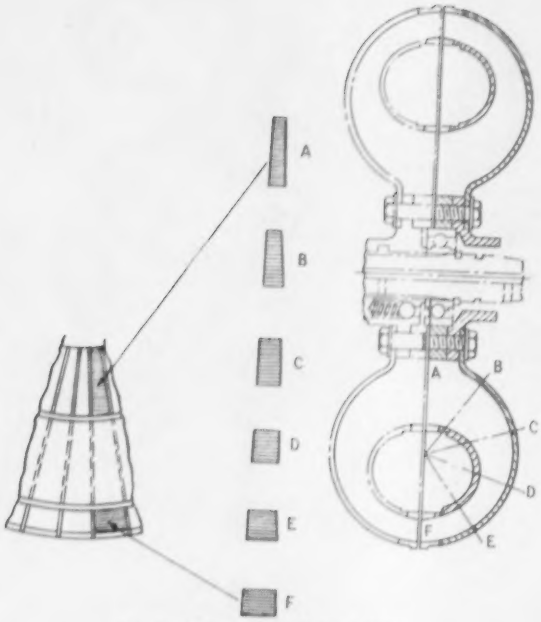


Fig. 4. Oil flow through torus ring

torque transmitting capacity of a hydraulic coupling must be high and the slip as low as possible. The slip is usually expressed as a percentage of the impeller speed, since it is the difference between the rotational velocity of the impeller and the runner. The amount of slip under full engine load determines the coupling efficiency. On the other hand, the torque transmitting capacity must be low when the engine is idling in gear. This is termed the *drag torque* of the coupling. Incidentally, high vehicle speed is usually obtained at engine revolutions at which torque is not at its maximum, so that under such conditions percentage slip would be negligible in relation to that for lower or normal vehicle speeds. The third requirement concerns vehicle acceleration, during which the change from engine idling with low drag torque to that of running condition with low slip must be effected without delay.

Research work by the Vulcan Sinclair Company over a period of years has established many of the fundamentals of the present day designs. Perhaps the most outstanding conclusions reached, with reference to the specific adaptation of the fluid flywheel to the automobile, may be stated as follows:

1. That as a means of reducing idling drag torque, although not affecting torque capacity of the coupling, a baffle may be fitted, but only of limited dimensions if the coupling be fitted with a core guide ring.
2. In a working circuit without a core guide ring, there is not the same limitation in baffle size. Performance is still further improved by the inclusion of a central reservoir of capacity in the order of one quarter of the fluid in the working circuit.
3. That with such a design, it is possible to combine an increased number of vanes, although the amount of filling may be reduced, in order to improve the idling drag torque characteristics.
4. If all these design points be combined, the volume of oil in the working circuit may be varied over a wider range, for adjustment of the full torque stalling speed whilst not impairing normal running speed slip.

Mention has been made of a baffle for anti-drag purposes. This usually takes the form of a circular plate attached to either rotor at its small diameter, but projecting inside the torus, to form an obstruction to the flow of the working fluid. The obstruction is more marked at idling speeds and under starting conditions, as the velocity of the vortex flow is then comparatively high when compared with the speed of the runner round its axis. Under such conditions, the centrifugal force in the vortex flow forces the fluid against the outer wall of the torus and it is then the baffle exerts its retarding

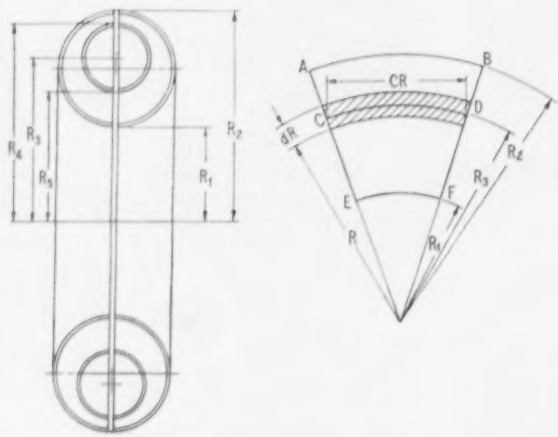
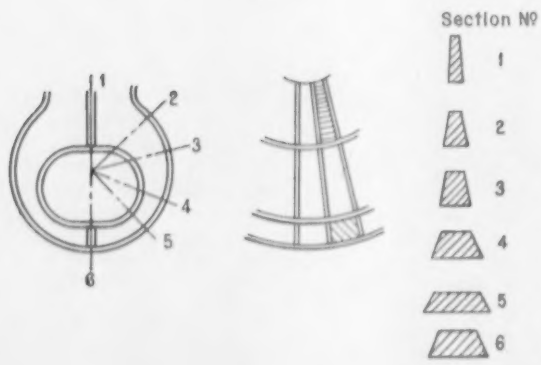


Fig. 6. Nomenclature adopted in the text for the determination of the sectional area of fluid flow between vanes

Fig. 5. Illustrating the constant area between torus vanes at different sections



influence. This effect is reduced as the speed of the runner increases, concurrent with which the centrifugal force due to vortex motion is decreasing. Owing to its cushioning effect, the baffle is advantageous in cases where the fluid coupling is used in conjunction with an epicyclic gearbox, insomuch as it has the effect of reducing shock emanating from a rapid change from high to low gear.

These are only a few of the fundamental principles of operation of the hydraulic coupling, and before proceeding to deal with the torque converter, certain formulae may be set down which will enable general overall dimensions to be ascertained as a preliminary to the more involved mathematics of detail design.

Previous comments have been made upon the effects of the quantity of fluid in the coupling and it is therefore desirable at this stage to consider this question with reference to initial calculations concerning output characteristics, and vane proportions. Figs. 4 and 5 illustrate a fluid coupling in which the impeller and runner form annular channels subdivided by radial partitions, forming cells. Since the cells are narrower near the coupling axis and wider at the periphery, the fluid will not have its centre of motion coincident with the geometric centre of the cross section of the impeller and runner. However, it is possible to determine this centre of rotation of flow from the fact that should the flow be unrestricted by interior walls, then the rate of flow will be similar from impeller to runner as it would be from runner to impeller. Therefore, the sectional area of flow will be constant at all points in its circuit in any one direction. Assume that in Fig. 6 *ABEF* is one of the cells, whose width is proportional circumferentially to its inner and outer distance from point *O*. The fluid circulates about the axis *CD*. Therefore, if C = constant, $AB = CR_3$, $EF = CR_1$, and $CD = CR_2$, the two areas of *ABCD* and *CDEF* must be equal, hence:

$$\frac{CR_3 + CR_1}{2}(R_3 - R_1) = \frac{CR_2 + CR_3}{2}(R_2 - R_3)$$

and $$R_3 = \sqrt{\frac{R_1^2 + R_2^2}{2}}$$

This, therefore, is the origin of fluid circulation throughout the cells. It is known that the flow through the impeller is widely divergent from that of the runner portion. This is because it travels along the outer walls of both sections and undergoes a greater change in velocity between entering and leaving the impeller section than does the remaining fluid, which flows at a smaller radius from the centre of circulation.

To facilitate calculation, therefore, it would appear necessary to determine an equivalent mean path of flow, which would transmit the same power as provided by the flow through the actual circuit at the same rate of flow and speed of rotation.

If in Fig. 6, a small element of fluid of thickness δR and width CR is entering the runner cell, then its rate of flow is proportional to its area $R\delta R$, the first R in the expression being the distance from the origin of fluid rotation. Its circumferential velocity is proportional to its distance from the axis while for each unit of time, the kinetic energy of the fluid will depend upon its mass and the square of its velocity, so that if E = kinetic energy, $\delta E = CR^3\delta R$. Integration between the two limits $R = R$ and $R = R_2$ determines the energy in the fluid over the whole inlet area. Thus:

$$E = \int_{R_2}^{R_3} CR^3\delta R = C\frac{R_3^4 - R_2^4}{4}$$

Its mean value per unit of radial thickness is expressed as:

$$C\left[\frac{R_3^4 - R_2^4}{4(R_3 - R_2)}\right]$$

and if this value be the same as that for the energy per unit flow in the actual case, assuming entry at radius R_4 when the energy would be CR_4^3, then:

$$CR_4^3 = C\left[\frac{R_3^4 - R_2^4}{4(R_3 - R_2)}\right]$$

and $$R_4 = \sqrt[3]{\frac{R_3^4 - R_2^4}{4(R_3 - R_2)}} \text{ in}$$

The mean radius of fluid entry into the runner is R_4, and if the mean radius of outlet is represented by R_5:

$$R_5 = \sqrt[3]{\frac{R_3^4 - R_1^4}{4(R_3 - R_1)}} \text{ in}$$

The velocity of the fluid at these equivalent radii, if N = speed of runner in r.p.m., will be:

$$\frac{N}{60}2\pi\frac{R_4}{12} \text{ ft/sec for in-flow radius}$$

and $$\frac{N}{60}2\pi\frac{R_5}{12} \text{ ft/sec for outlet radius}$$

Thus, the mean change in velocity from inlet to outlet will be:

$$\frac{N}{60}2\pi\frac{R_4 - R_5}{12} \text{ ft/sec}$$

for a vane pressure per unit per sec of

$$\frac{1}{g}\frac{N}{60}2\pi\frac{R_4 - R_5}{12} \text{ lb}$$

and since this load is exerted on the vanes at the mean radius, the produced torque for each unit of fluid is:

$$\frac{1}{g}\frac{N}{60}2\pi\frac{R_4 - R_5}{12}\frac{R_4 + R_5}{24} \text{ lb-ft}$$

This can be simplified to:

$$\frac{N(R_4^2 - R_5^2)}{88{,}450} \text{ lb-ft}$$

The total torque on the runner, using previous notation, may be expressed for any weight of fluid:

$$T = \frac{WN(R_4^2 - R_5^2)}{88{,}450} \text{ lb-ft}$$

Substituting this expression in that for the power output, which can be stated as $HP = 2\pi NT/33{,}000$,

$$HP = \frac{2\pi NT}{33{,}000}$$

$$HP = \frac{WN^2(R_4^2 - R_5^2)}{465{,}000{,}000}$$

with a fluid circulation of:

$$W = \frac{465{,}000{,}000}{N^2(R_4^2 - R_5^2)} \text{ lb/sec}$$

This expression is in terms of fluid weight and to ascertain the actual volume, it is necessary to substitute as follows:

$$(R_2 - R_1)\frac{2\pi}{4}\left(\frac{R_2 + R_1}{2}\right)2\pi = \frac{\pi^2}{4}\left[(R_2^3 + R_1^3) - (R_2R_1^2) - (R_1R_2^2)\right] \text{ in}^3$$

This assumes no reduction in volume of contained fluid in the coupling due to thickness of vanes, and accordingly some adjustment must be made to ascertain the actual effective volume. The foregoing formulae will be useful in obtaining overall dimensions of a preliminary nature.

Deceleration takes place in the runner oil, the change in velocity possessing a tangential component which represents the expenditure of kinetic energy in turning the runner. The power output is less than the input power as also the runner speed is less occasioned by slip, which is usually between 2 and 4 per cent, with a coupling efficiency of approximately 96 to 98 per cent.

Of course, the transmission of the power from the impeller to the turbine depends upon the liquids circulating under a certain pressure. The greater the pressure created by the impeller for a given output, the smaller will be the quantity of the liquid necessary for given power transmission values. On the other hand, the amount of liquid required is greater in inverse proportion to the selected pressure. Blade shape, quantity of fluid, and pressure together, therefore, determine the character of the torque curve. By suitable compromise between these three features, the starting characteristics can be controlled and here it should be stated that the greater the amount of fluid in proportion to the head, the more favourable will the starting torque become. By means of such compromise, the *secondary torque* can be also increased by increasing the number of turbine stages in the converter

or alternatively the same starting torque can be achieved with a single stage converter having reduced pressures, by comparison with the two-stage unit.

There are, of course, inherent losses in any converter, as there are in an internal combustion engine, most of which occur through the blading and eddy current losses in the former. The principal losses of normal character comprise:

1. Friction loss in pump or impeller wheel, approximately 4·5 per cent.
2. Friction loss in turbine blading, approximately 3 per cent.
3. Mechanical losses, 1·5 per cent.
4. Leakage losses, 3·5 per cent.

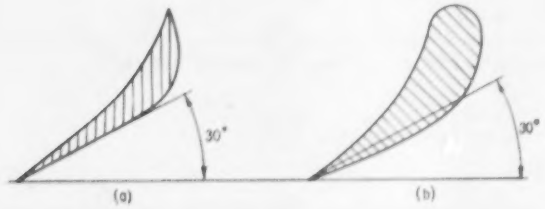


Fig. 7. Blade forms. (a) experimental reaction blade design ; (b) experimental blade design for varying inlet angles

It has been stated that the efficiency of a hydraulic gear of the turbine type is in the order of 90 per cent. Of the friction and build-up losses which occur, those of blade friction loss are among the highest, excluding those in the pump wheel. They are certainly of paramount importance in the case of a closed circuit turbine type converter, which is required to operate at widely different speeds of input and output shafts, and at varying torque ranges. These losses are critical, particularly at starting and secondary speeds where, as previously explained, the input and output torques are equal or where the output falls below the input torque value.

Theory of blade friction

A brief outline of the theory of blade friction is therefore necessary although, as will be appreciated, this cannot here be treated completely, due to space limitation. In accordance with the principles of hydrodynamics, the curve representing the efficiency at different speeds of the turbine shaft, under constant energy supply will be in the form of a parabola. This applies if the turbine consists of one runner only, when the output energy cannot be utilized. If, however, this energy is utilized during its stages from one wheel to another, then the frictional efficiency varies differently. Assuming the turbine is of the axial flow type and possesses an infinite number of stages, then the drop in pressure in the blade will be:

$$\Delta p_u = \frac{V_d^2 - V_1^2}{2_g}$$

and the drop on account of friction:

$$\Delta p_f = \frac{f}{2}\frac{V_d^2 + V_1^2}{2_g}$$

where V_d = the discharge velocity from the blade, V_1 = the inlet velocity to the blade, and f = the friction coefficient. The total pressure drop will, of course, be the summation of the two expressions:

$$\Delta p_u + \Delta p_f$$

$$\text{or,}\quad \frac{V_d^2 - V_1^2}{2_g} + \frac{f}{2}\frac{V_d^2 + V_1^2}{2_g}$$

from which the efficiency of each blade may be obtained as:

$$E = \frac{\Delta p_u}{p}$$

Consider the efficiency at different speeds. If the energy supply be constant, then its product with the volume of circulating fluid must also be constant, or mathematically expressed as $K = C\Sigma\Delta p$, where C = the volume of fluid.

More recent research, however, disposes of some of these earlier conclusions, amongst them the belief that in order that the hydro-kinetic transmitter may effectively perform the duties of a clutch, the characteristics of the vanes of the driven element must of necessity differ from those which would be necessary to perform the functions of a variable gear, and consequently that to combine the two functions in a single transmitter could not be achieved without the adoption of either variable or adjustable blades. It was a so considered that only within a comparatively small range of variation in fluid velocity could the power transmitter function with reasonable efficiency. Furthermore, the functions of the clutch and variable ratio gear, being so widely different, in the case of the automobile, rendered the efficient performance of the single transmitter almost impossible.

It is now believed that if the transmitter blade is profiled and disposed such that the fluid action, when in combination with the action of the reaction element, varies with the varying conditions, no such adjustment or variation in profile of the blade is necessary. This may be achieved by careful blading and ducting, the design of which reduces the possibility of unsteady fluid motion within the ducts of the hydraulic circuit and which in consequence eliminates any shock at the inlet to the driven element.

At the same time, it was considered that the best speed: efficiency characteristics were obtained by open reaction blading of profile similar to (*b*) in Fig. 7, that is the blunt, rounded inlet edge, although the energy circulation together with efficiency may be reduced considerably by fluid surge at low secondary and stalling speeds if the blades were too thin and open. The efficiency curve of the turbine could be displaced to either side of the optimum line of speed in two

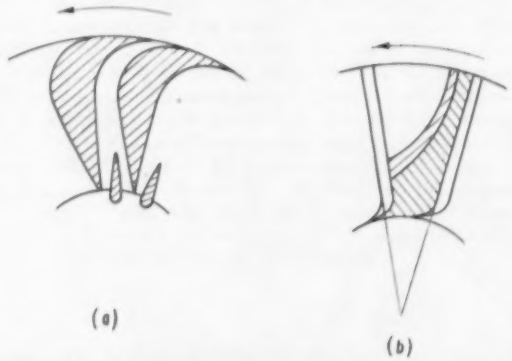


Fig. 8. (a) layout of vanes to control fluid velocity and to reduce flow disturbance ; (b) showing the path of fluid flow along the trailing edges of turbine vanes

ways. The first is by the use of a partial impulse type of blade which lowers the efficient speed, and the second is the use of open and flat type of reaction blade in order to displace the curve to a higher speed level. Later developments in blading and ducting show that positive control of fluid motion is not possible and it is therefore desirable at this stage to recapitulate the essential principles involved in

reducing any unsteadiness in fluid motion throughout the circuit, which is one of the pre-requisites of successful design.

It is known that the point at which any disturbance in flow is most detrimental is that at which the stream from the driving element ducts merge into an annular circuit which must bridge the gap between the two driving and driven elements, and with any uncontrolled eddy motion at this point, control of rotational velocity is rendered extremely difficult. A method of blading and a design of ducts which overcomes this difficulty can be laid out by disposing the vanes of the driving element in radial planes from their inner periphery to a point inside the major radius of the circuit, thence curving rearwards to the delivery ends, as in Fig. 8(*a*).

The necessary decrease in cross sectional area is obtained through the converging disposition of the faces forming the axial depth of the ducts whilst the sweep backwards of the vane enables the profile to be gradually reduced in section until finally reaching a fine edge at the delivery end. It will be seen that this reduces much of the inclination towards angular divergence of adjacent vane faces at the delivery end, a feature which, if allowed to exist, produces an unsteady flow of fluid. The proportions of the vanes are such that the receiving ends, being bulbous or rounded, bear appropriate relation to their pitch. In a radial inward flow turbine, the ducts between the vanes are divergent in order to produce a steady flow at high velocities, although much care must be exercised to ensure that angular divergence of opposite faces does not occur. This can be arranged if, say, in a rectangular cross section one opposite pair of faces is parallel and the opposite pair convergent.

Conditions are somewhat different in a wheel in which the flow is radially outward. The faces of the duct which

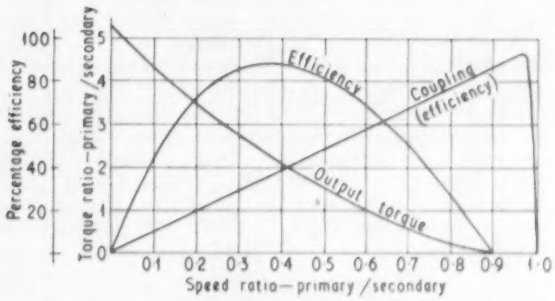


Fig. 9. Torque and efficiency curves against speed ratio of converter. The hydraulic coupling efficiency curve is also shown

were parallel in the inward flow turbine are now divergent, but with this comes the inherent disadavantage of instability of fluid flow, whether subjected to angular motion or otherwise. If angular motion at high speed is present, the tendency of the flow within the ducts is to follow a path along the trailing faces, leaving inert fluid at the leading faces, Fig. 8(*b*).

Hence, it is necessary to dispose the vanes and ducts of the driving element as shown in Fig. 8(*a*), in order that the fluid can travel in a uniformly steady stream between the driving and driven elements of the transmitter. Moreover, with such an arrangement, it will be seen that the radial portion of the ducts can be made to converge to the desired extent not only in their widths but also in their depths, whilst since the curved portion begins at approximately the radius of duct outlet, the fluid is deflected rearwards and decelerated in an angular direction in relation to the ducts. The angular and rearward disposition of the blade outlet end also imparts the necessary rearward velocity component, which varies approximately as the relative rotational speed of the driving and driven elements. It is probable that further research will reveal data regarding blade and duct design which will enable flow characteristics to be more accurately forecast during initial design.

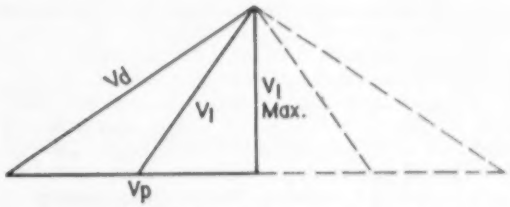


Fig. 10. Velocity triangles for constant liquid flow.

Reverting to the previously mentioned characteristic of overall efficiency, Fig. 9 shows that the torque curve of the turbine intersects that of the primary torque, and consequently the output torque becomes smaller than the input torque as the turbine speed is increased. This occurs during a stage when efficiency is falling, and indicates that little purpose would be served in utilizing the hydraulic drive beyond the point at which primary and secondary torques are identical, and therefore some means of obtaining directional drive is indicated when this stage is reached. Several methods of achieving this objective will be noted in the examples which follow later. Mechanically, there are several important points to be observed, mainly in connection with the operating fluid. The design must not permit short circuiting the fluid between the casing and the revolving parts, while some means must also be provided for the

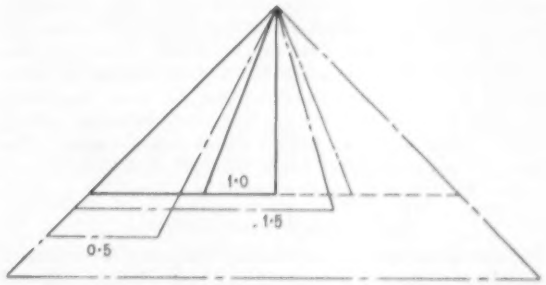


Fig. 11. Corrected velocity triangle, shown chain dotted, for intermittent speeds

evacuation of the gases emanating from the effects of temperature upon the circulating oil. In addition, the quantity of fluid must be maintained or cavitation may ensue with immediate loss of efficiency. The temperature of the circulating fluid should be controlled, preferably by the insertion of an oil cooler in the fluid circuit. In numerous designs from home and overseas, an injector is incorporated to feed the necessary fluid to the converter and to control its pressure, circulation being obtained from the converter pump.

It must be appreciated that, at increased flow as against constant flow, the pressure drop varies as some power of the flow and thus the energy consumption also varies with the velocity of flow. Thus, the velocity diagram, Fig. 10 must be adjusted to intermittent speeds between starting and racing. Fig. 11 shows these triangles modified for

speeds in relation to speeds of turbine at optimum velocity. The efficiency curve for a single-stage turbine is represented by a parabola drawn on a speed efficiency basis.

It has been mentioned that torque capacity of any fluid pump varies as the square of the speed and also that the power varies as the cube of the speed. This is, however, true only in a relative sense, as again is the assumption that the capacity horse power of a pump varies as the fifth power of the diameter. For sizes of hydraulic coupling above, say, 18–20in circuit diameter, these assumptions are moderately

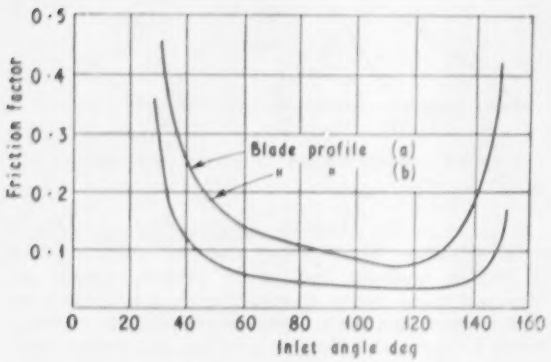


Fig. 12. Test results of friction factors for (a) blade profile (a) Fig. 7 and (b) blade profile (b) Fig. 7

correct, always providing that the circuits of the different designs are similar. In assessing relative performances of different diameter couplings of similar design, due cognisance of this fact must be appreciated, as below the diameter of 18in actual torque transmitted is appreciably below the amount expected from the original assumption. Performance relative to diameter is assessed by applying or comparing the performance by a known diameter which gives three per cent slip and five per cent slip at normal speed and the original diameter is corrected by the relationships of speed squared N^2 and diameter to fifth power D^5, to an equivalent coupling of, say, 10in, as a convenient diameter, having single circuit and operating at, say, 1,000 r.p.m. The corrected torque capacity would be stated thus:

$$T\,cor = T\left(\frac{1{,}000}{r.p.m.\ test}\right)^2\left(\frac{10}{unit\ diameter}\right)^5$$

Experiments carried out during the early design of a successful torque converter used on heavy vehicles indicated that the blade friction closely approximated the loss experienced in a smooth tube. Of many blade profiles tested, two are here exemplified with regard to their friction qualities, but first it must be stated that in order to obtain a flat efficiency curve the friction factor must be as low as possible, controlled to a major degree by the careful profiling of the blade. The two blade sections are both 30 deg angles, as shown in Fig. 7, but with widely differing profiles. The first, (*a*) is a reaction blade and (*b*) is one suitable for varying inlet angles. Tests on a complete converter equipped mainly with blades of the latter section, showed that the friction factors varied between 0·025 at 130 deg inlet angle to 0·1 at 50 deg inlet angle, and efficiency at 1,000 r.p.m. primary was 86 per cent, increasing to 89 per cent at 1,400 r.p.m., Figs. 12 and 13. Consumption of power decreases from its maximum at stalling speed in approximately a straight line curve. This will be seen from Fig. 13, in which at 1,400 r.p.m. racing, the primary input has been reduced by 60 per cent, leaving only 40 per cent of the input as at stalling speed.

Earlier tests indicated that adjustable pump blades were more efficient than the set angle blades and that they possessed several useful characteristics. In view of later experience, however, this is somewhat dubious.

A velocity triangle can be drawn, as in Fig. 10, from which it will be seen that the value of V_i decreases to a minimum until a point is reached where it becomes perpendicular to Vp. The triangle is then repeated on the other side of its perpendicular and V_i increases until it reaches V_d at racing speed.

The value of p_u varies between zero and a maximum, and p_f between $(f/2)(V_d^2/g)$ and a minimum. The total pressure drop varies between $(f/2)(V_d^2/g)$ and a maximum, as has been seen from the previous expressions. Therefore, all the available energy is absorbed by friction at points of maximum torque and stalling, and no torque at racing speeds. The flow of fluid in relation to the flow at $V_{i\,max}$ must, however, be increased at all speeds if $C\Sigma\Delta p$ is to be retained as a constant. It has been stated previously that the pressure drop varies with the second power of the flow with any increase, and that the consumption of energy varies with the third power of the flow velocity, so that in relation to $V_{i\,max}$, the pressure drop Δp may be expressed as:

$$\sqrt[3]{\frac{\Delta p\ max}{\Delta p}}$$

This increased flow must be incorporated in the velocity ratio triangles, Fig. 11, in the form of flow at intermediate speeds and also at starting and racing speeds. These

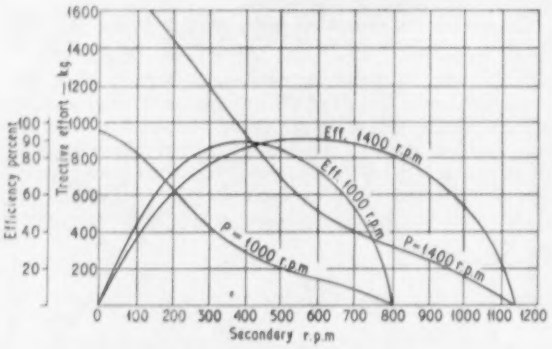


Fig. 13. Tractive effort-efficiency curves

speeds are relative, that is, turbine speed in relation to optimum turbine speed. The ensuing diagram denotes the *corrected* energy consumption values.

The smaller couplings experience the largest decrease in capacity when subjected to this correction, and since the drag torque is not affected to such an extent the ratio of drag torque capacity deteriorates rapidly in proportion to the decrease in coupling circuit diameter. It is therefore possible to utilise the drag torque factor as one other basis of comparison. This factor is represented as a ratio between drag torque or torque at stalling point and the transmitted torque at 3 per cent slip, the speeds for each being common, or in order to avoid high test speeds the drag torque at normal operating stall speed is corrected by the square law until the speed is reached, whereupon the 3 per cent slip is obtained. The factor may be written as:

$$TDf = \frac{TD\left(\frac{N_1}{N_2}\right)^2}{Ts}$$

where TDf = drag torque factor, TD = drag torque at normal stall speed, Ts = torque capacity at 3 per cent slip

at normal operating speed, N_1 = normal operating speed and N_2 = normal stall speed.

Torque converter

The torque converter differs from the fluid flywheel or hydraulic coupling insomuch as it comprises three essential units instead of two. These are the impeller, the driven member and a reaction member. All three members carry curved vanes in place of the radial ones in the hydraulic coupling. When the engine is started up, the impeller vanes impart motion to the fluid and thus kinetic energy is produced. Change in direction of fluid flow occurs as the oil passes into the vanes of the driven member, sometimes called the *turbine member*, producing motion in the second member, since some of the kinetic energy is absorbed by the turbine during fluid flow diversion.

From this state, the stationary reaction member redirects the fluid to return to the impeller vanes, the profile of the reactionary vanes being such as to carry out this function so that the re-entrant is at the best angle to produce acceleration of the impeller. Both profile and angles of these vanes are critical in the design of the turbine since they control the degree of torque conversion. The efficiency of the hydrokinetic converter is in the order of 90 per cent, depending upon the blade forms and the number of stages in the impeller and runner, whereas the plain hydraulic coupling attains an efficiency of as high as 98 per cent, due in the main to the higher degree of turbulence in the converter created by the several changes in direction of the fluid flow. Moreover, the two members of the coupling, the impeller and runner, are rotating during steady operation at approximately equal speeds, a condition which is but rarely encountered in the converter, since the stationary reaction member provides obstruction to the fluid flow, resulting in higher vortex and friction losses. The efficiency is zero at both starting and at the period when the runner is allowed to idle, for since in the latter condition there is no torque, there can be no output. The output torque is a maximum when starting, dropping below the input torque at approximately 63 per cent of the input speed, when the two torque values are equal, there being at that instant no reaction on the guide vanes. However, as the output falls lower in relation to the input torque, the moment on these vanes is reversed in direction, and they rotate in the same direction as the impeller and runner. These conditions are shown graphically in Fig. 14.

It will further be appreciated from the curves in Fig. 14 that at starting speed the torque is multiplied considerably, but as speed increases, this factor decreases rapidly until the output shaft is rotating at 40 per cent of that of the input shaft speed when the efficiency is highest at approximately 85 per cent. Further increase in vehicle speed reduces the efficiency of the converter. An improvement in efficiency can be obtained by using a hydraulic coupling in combination. If the efficiency curve of a coupling be superimposed as on Fig. 9, it will be realised that a change-over from torque converter to coupling will be beneficial at a point where the two efficiency lines coincide, approximately 64 per cent in the graph.

The function of such a torque converter, therefore, would be primarily for acceleration purposes throughout the lower speed range of the vehicle, while when the speed is such that the engine can drive the vehicle in direct gear, it should be coupled direct to the transmission line by hydraulic coupling or, as in some designs, by a friction clutch.

In the design of a torque converter to be efficient and mechanically sound a few points must be observed, amongst which the major ones are as follows:

1. The prevention of short circuit of fluid between casing and revolving parts.
2. Prevention of fluid loss.
3. The expulsion of gas from the converter, as new fluid is likely to gasify slightly under certain conditions. Generally preventive measures draw the gas through a small pipe from the top of the converter to an ejector.
4. Prevention of fluid cavitation and provision to ensure full fluid pressure by an injector from a reserve tank.
5. Provision for temperature rise by means of an auxiliary oil cooler in the circulation system. The cooler should include a fine oil filter.
6. To bleed the converter casing, preferably from two points each at different radius, so that when the converter is running, the pipe from the casing to the casing top carries fluid under pressure. Thus, a vacuum is created at the ejector, which ensures the drain sump is empty. This pipe also acts as an expansion pipe in addition to its being an exit for gases. The injector pressure is controlled by an internal, spring-loaded ball valve.

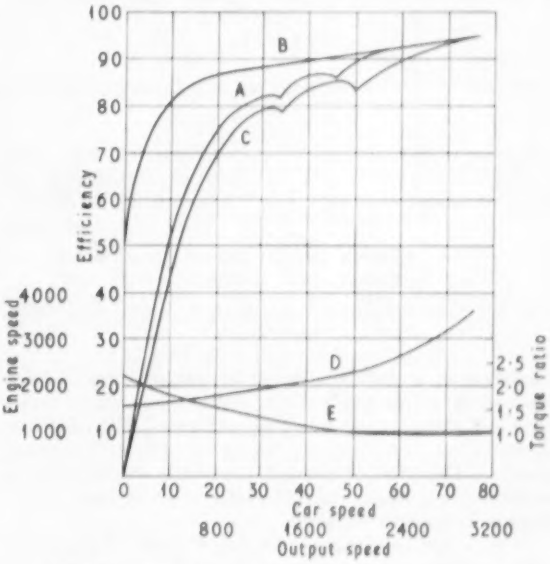


Fig. 14. Typical curves for engine-torque converter combination

In order to obtain the best performance from a converter, it should be used with a prime mover or engine of suitable torque:speed characteristics. The easiest way to alter these characteristics is, where possible, to:

(*a*) Reduce the diameter of the pump wheel if it is desired to use reduced engine torque, or

(*b*) Increase the effective outlet angle of the pump blades for higher torque or by increasing the number of blades.

(*c*) To introduce a primary step-up or reduction gearing in the transmission.

To reiterate, the attainable efficiency depends to a large extent upon the design of blading, and a study of the flow through the blades shows that the influence is to be considered over the whole area of the shape or profile, but with emphasis on the conditions at inlet and outlet of the blade face.

The optimum efficiency in the impeller and guide wheels is only obtainable at one single point on the curve, which can be arranged to be high and reasonably flat over quite a wide range by suitable blade design for rotational and stationary wheels.

The effects of variations in pump diameters and blade angles are critical. For instance, if the diameter be reduced,

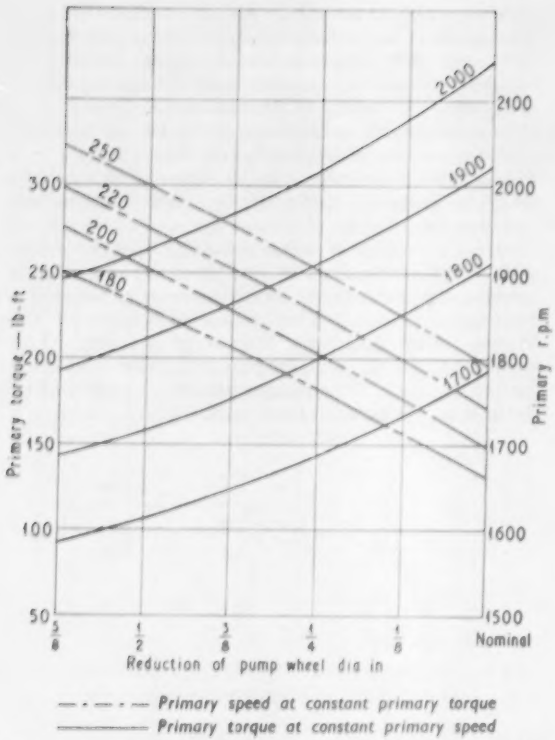


Fig. 15. Speed and primary torque curves for varying pump wheel diameters

the efficiency at the lower end of the efficiency curve is also reduced, while the whole curve, due to the increased input, is moved along the secondary speed range. The efficiency, therefore, is comparatively higher at greater secondary speeds. The effect of reduced diameter is also apparent in stalling torque ratio, which increases in about inverse proportion to diameter variation.

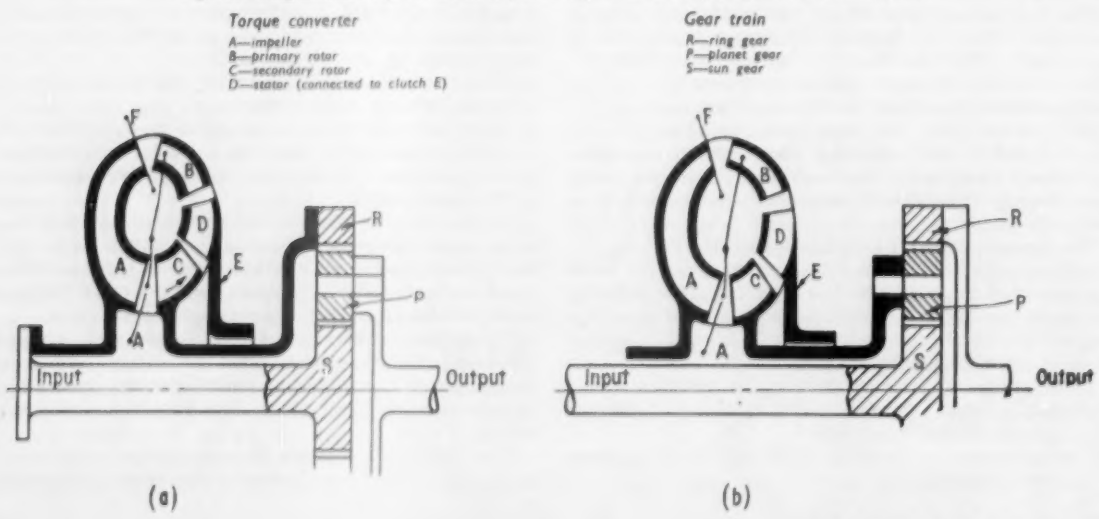
Typical speed and primary torque curves for varying pump wheel diameters for a constant secondary speed of 1,000 r.p.m. are illustrated in Fig. 15. Amongst other factors which determine the speed at which maximum turbine efficiency is obtained, there are (*a*) the number of stages, (*b*) the blade ring dimensions and (*c*) the head to be adopted. This speed can be increased by increasing the head on the turbine side, without modification to the blade length but by reducing the blade angle, and on the pump side by altering either or both the blade angle and wheel diameter. These few preliminaries should enable a clear conception to be obtained of the functions of, and the principles governing both the fluid coupling and the converter, prior to a brief examination of the method of driving the vehicle by direct drive.

The necessity for a gear train behind the converter has been mentioned, and it is interesting to note the adoption of epicyclic gearing in this connection. There are, however, several points to be taken into consideration in this respect, and it is, therefore, advisable to bear in mind the peculiarities of certain systems, insofar as they may affect the desired output.

Some of the systems have been discussed by A. Y. Dodge of U.S.A., in *Automotive Industries*, in which it is stated that three classes embrace the different types for most general purposes, and here it must be stated that although an epicyclic train may be placed in one class, it does not necessarily follow that the same train cannot function in one or other of the remaining categories. There is the straight mechanical reduction in which one element is fixed, and also the regenerative and non-regenerative types. The regenerative type requires that when the three elements of the gear are in motion, a proportion of the power transmitted returns to the driving shaft through a differential. With the non-regenerative system, under similar conditions of rotation as for the regenerative type, the connections of the elements is such that no return of power to the driving shaft is possible although it is transmitted in two distinct directions, the total output being equal to the input power. The last two categories, non-regenerative and regenerative transmissions, are illustrated in Fig. 16.

Regenerative circuits are not always desirable as they tend to reduce overall efficiency and when compared with the non-regenerative type, the efficiency is nearly always less. Moreover, some of the components within the circuit may be caused to transmit more power than the magnitude of either input or output power. It is possible, however, to design the regenerative system to provide a large reduction with very few gears in a comparatively small space. Hence, the arrangements that have from time to time been produced, in which a regenerative differential planetary gear is designed to revert to the non-regenerative type. Such an arrangement posses the advantage that it is possible to obtain, even with

Fig. 16. Diagrammatic representation of (a) non-regenerative and (b) regenerative transmission



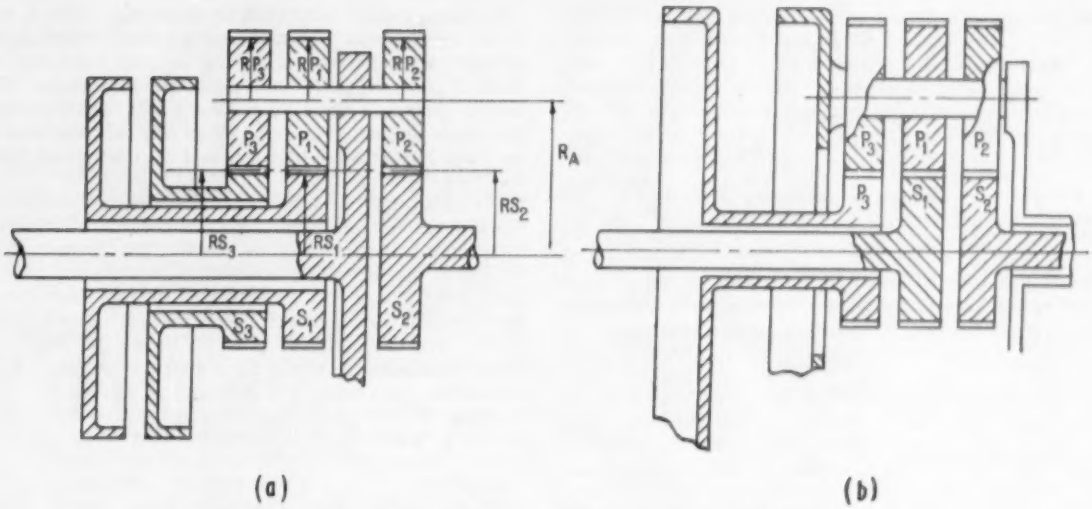


Fig. 17. Epicyclic gear trains ; (a) case driving and driven ; (b) sun gears driving and driven

a smaller number of gears, any desirable change in leverage and planetary characteristics.

In the non-regenerative system it will be seen that the impeller of the torque converter is connected to and rotated with the sun wheel of the gear train, whilst the ring gear of the transmission turns with the rotor. In the regenerative type, the rotors are connected to the planet gears and the ring gear to the output shafts.

In Fig. 16a, the two-way power flow reduces the increase in output torque at starting to a torque only slightly below that of the directly-connected torque converter. The gears, however, cause the rotor to turn in the reverse direction at starting and thus the torque ratio of the rotor to the impeller is increased during this starting period.

In cases where there is no reaction member, such as obtains in the fluid flywheel or plain hydraulic coupling, the torque input will obviously be equal to the output torque, excluding the losses due to friction in internal mechanism, and in the event of a two-directional power flow, some of the torque is transmitted through the gears and the remainder through the fluid flow. It will be seen that in this way it is possible to reduce the dimensions of the coupling since the hydraulic units will be relieved of a portion of the power which they otherwise would transmit if a complete hydraulic system were adopted. A regenerative planetary gear system differs somewhat in these respects, since the torque load imposed upon the rotor exceeds both that at input and output, whilst the power passing through the hydraulic circuit also exceeds these torques. In the non-regenerative type, the efficiency at all points in the circuit is higher than that of the hydraulic efficiency, but the opposite applies in the regenerative system. The tooth contact line in epicyclic gearing, since it is not radial, produces a torque loss on account of the gear tooth friction, and the gear efficiency is in consequence reduced. Although the presence of this friction assists in torque transmission from driving to driven shaft, it will be appreciated that the torque per cent transmitted by gear friction fails to meet with torque increase. In other words, if the torque ratio be 1·5:1 and, say, 5 per cent were transmitted through gear friction, this would mean that for every 100 lb-ft input torque the output would be 95 lb-ft. The torque losses are, however, small and in order to avoid unnecessary complication in calculations of torque transmission it is sufficiently accurate to assume 100 per cent gear efficiency, making any adjustment in this respect at a subsequent stage. The following formulae deal with tooth loads and torques for both the regenerative and non-regenerative systems. In Fig. 17*a*, where ME = mechanical efficiency and R and S are the ring and sun gear diameters:

$$ME = \frac{R}{S} + 1$$

and the revolutions of the ring gear, arm or cage and sun gear are respectively: R_R, A_R and S_R

$$S_R = \left[A_R\left(\frac{R}{S} + 1\right)\right] - \left(\frac{R}{S}\right)R_R$$

$$A_R = \frac{\left[\left(\frac{R}{S}\right)R_R\right] + S_R}{\frac{R}{S} + 1}$$

$$R_R = \frac{\left[A_R\left(\frac{R}{S} + 1\right)\right] - S_R}{\frac{R}{S}}$$

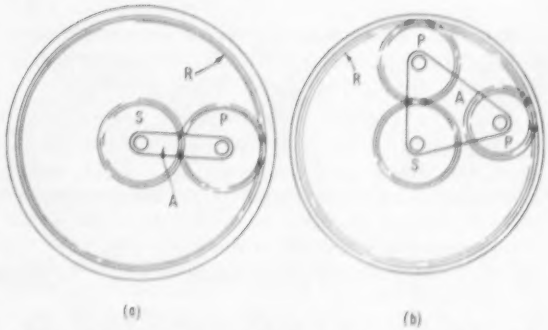


Fig. 18. Systems of epicyclic gearing for regenerative and non-regenerative sets

In Fig. 17*b*, with similar notation:

$$ME = \frac{R}{S}$$

and

$$S_R = \left[R_R - \left\{\left(1 - \frac{S}{R}\right)A_R\right\}\right]\frac{R}{S}$$

$$A_R = \frac{R_R - \left(\frac{S}{R} \times S_R\right)}{1 - \frac{S}{R}}$$

$$R_R = \left(\frac{S}{R} \times S_R\right) + \left(1 - \frac{S}{R}\right)A_R$$

For the non-regenerative conditions of both Figs. 18*a* and 18*b*, the following conditions are applicable:

1. Cage or arm is the output or driven member.
2. Sun wheel receives input from engine.
3. Ring gear receives input from variable speed device.

Then, the formulae for unit torques may be stated.

$$\textit{Rotor torque} = \textit{Ring torque}$$

$$\textit{Sun wheel torque} = \frac{\textit{Rotor torque}}{ME - 1}$$

$$\textit{Sun wheel torque} = \frac{\textit{Output torque}}{ME}$$

$$\textit{Impellor torque} = \textit{Sun torque}\ (ME - 1)\ \frac{\textit{Torque of driven member (variable device)}}{\textit{Torque of driving member (variable device)}}$$

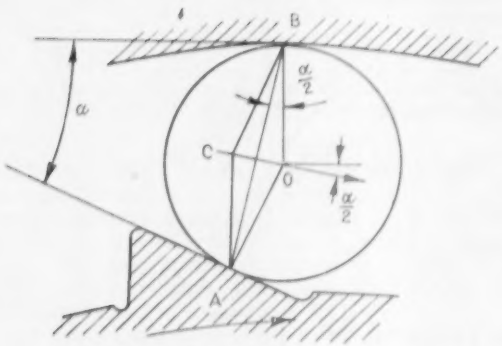


Fig. 19. Over-running clutch (roller type)

The regenerative conditions for Figs. 18*a* and 18*b* are:

1. Cage or arm receives input from variable speed device.
2. Ring gear becomes the output or driven member.
3. Sun wheel; reaction driven, direction forward, since it imparts effort to the impeller its torque is negative.

Then, the arm or cage torque $= (L_1 \pm L_2)A$, and the following expressions obtain under the various conditions of drive.

Torque output:

(*a*) When sun wheel 1 is driven, torque $= RS_1 \times LT$ where LT = tooth load.

(*b*) When arm or cage is driven, torque $= R_A(L_1 \pm L_2)$.

(*c*) When sun wheel 3 is fixed and the cage driven,

$$\text{torque} = R_A\left(1 \pm \frac{RP_1}{RP_3}\right)$$

Torque input:

(*a*) When sun wheel 1 is driving, torque $= RS_1 \times LT$.

(*b*) When cage is driving, torque $= RA(L_1 \pm L_2)$.

(*c*) When sun wheel 3 is fixed and the cage driving,

$$\text{torque} = R_A\left(1 \pm \frac{RP_1}{RP_3}\right)$$

Assuming that the tooth loads on the output wheels S_2 and P_2 are equal to unity, then the tooth load on wheel $P_3 = RP_2/RP_3$ when the arm is driving and the tooth load on wheel $P_1 = RP_2/RP_1$ under similar conditions. The reaction torque on wheel $S_3 = RP_2 \times RS_2/RP_3$, also when the arm is driving. Note that if the original tooth load in the foregoing formulae be taken as 1 lb, the various ratios may be secured.

For calculation of the speeds of revolution of wheels and pinions, the following will form the nucleus upon which each individual speed may be evolved, for instance, when S_2 and S_1 are the wheels involved, S_2 and P_2 will be substituted for S_2 and P_2. Furthermore, the cage and two sun gear formulae can be adopted for all three sun gears turning or one stationary. If the prefix V before the wheel and pinion members represents the velocity in rev/min of the component, then when S is stationary in all cases:

$$\frac{VS_1}{VA} = 1 - \frac{RP_1 \times RS_2}{RP_2 \times RS_1}$$

$$VA = \frac{VS_1}{\left(1 - \frac{RP_1 \times RS_2}{RP_2 \times RS_1}\right)}$$

$$VS_1 = VA\left(1 - \frac{RP_1 \times RS_2}{RP_2 \times RS_1}\right)$$

The directions of revolution of A and S_1 are similar for positive values and opposite for negative values of these expressions:

$$VP_1 = \frac{RS_1}{RP_1}(VS_1 - VA)$$

However, if $VS_1 = 0$, then:

$$VP_1 = VA\left(1 + \frac{RS}{RP}\right)$$

$$VA = \frac{VS_1}{\left(1 - \frac{RP_1 \times RS_2}{RP_2 \times RS_1}\right)} + \frac{VS_2}{\left(1 - \frac{RP_2 \times RS_1}{RP_1 \times RS_2}\right)}$$

$$VS_2 = \left[VA - \frac{VS_1}{\left(1 - \frac{RP_1 \times RS_2}{RP_2 \times RS_1}\right)}\right] \times \left(1 - \frac{RP_2 \times RS_1}{RP_1 \times RS_2}\right)$$

$$VS_1 = \left[VA - \frac{VS_2}{\left(1 - \frac{RP_2 \times RS_1}{RP_1 \times RS_2}\right)}\right] \times \left(1 - \frac{RP_1 \times RS_2}{RP_2 \times RS_1}\right)$$

Taking the conditions with sun gears driving and driven as in Fig. 17*b* and using similar notation, then:

Torque output:

(*a*) When sun is driven, torque = driven sun $\times$ LT.

(*b*) Driven arm, torque = driven arm $(L_1 \pm L_2)$.

(*c*) When S_2 is fixed and S_1 is driving,

$$\text{torque} = RA\left(1 \pm \frac{RP_1}{RP_2}\right)$$

In these cases also the values are positive for similar rotation direction of A and S_1 and negative when these directions are opposed.

For revolution speeds:

$$VS_2 + \left[\frac{RS_2}{RS_1}\left(-\frac{RS_2 - RS_1}{RS_2 - RS_3}\right)\right] \pm \left[\frac{RS_2}{RS_3}\left(-\frac{RS_2 - RS_1}{RS_1 - RS_3}\right)\right]$$

$$VS_1 = \left[VS_2 \pm \frac{VS_3}{\frac{RS_2}{RS_3} \pm \left(-\frac{RS_2 - RS_1}{RS_1 - RS_3}\right)}\right] \times \left[\frac{RS_2}{RS_1}\left(-\frac{RS_2 - RS_1}{RS_2 - RS_3}\right)\right]$$

$$VS_2 = \left[\pm VS_2 \pm \frac{VS_1}{\frac{RS_2}{RS_1}\left(- \frac{RS_3 - RS_1}{RS_2 - RS_3} \right)} \right] \times \left[\frac{RS_2}{RS_2}\left(- \frac{RS_2 - RS_1}{RS_1 - RS_2} \right) \right]$$

Before concluding by generalities, the origin of investigations that led to the present stage of fluid or hydraulic torque conversion for automobiles should be considered. This will assist in presenting the problems involved more clearly, and at the same time illustrate some means of overcoming them.

It will be noted that in some of the transmission systems at present in use, an over-running clutch is included in the mechanism. Such a clutch permits motion to be transmitted through itself in one direction only. Its application may generally be explained as follows.

If one shaft drives another shaft through a train of gears, and one of the gears incorporates an over-running clutch, a second gear train can be employed to give a higher speed ratio, because when the second train takes up the drive, the clutch operates and the first train begins to idle. A direct drive may be used instead of the second train if desired, provided that the first train supplies the necessary gear speed reduction. There are many devices which will perform this function, but the type most extensively used in automobile transmissions is the roller type, which resembles a roller bearing insofar as it comprises an inner and outer ring, housing a series of rollers, (see Fig. 19).

One of the rings in the over-running clutch is formed with a series of cam-or wedge-shaped surfaces, each roller making contact with one cam face. A wedge action is therefore occasioned between roller and face upon movement of either ring. This angle of wedge formed by the tangent at the point of contact between the roller and cam surface in relation to the contact point of the roller with the cylindrical surface, is of paramount importance, as it must not exceed twice the friction angle. If such a condition does exist and the cam face exceeds twice the friction angle, then the roller will probably be squeezed out and the clutch will not function whilst it will also be appreciated that the stress in the rings depends upon this angle, together with the friction coefficient of the materials employed. It is appropriate, therefore, to set down the geometry of the clutch rollers and cam faces.

Referring to Fig. 19, assume that the rotation of the inner ring is clockwise, in which case the cam contacts the roller at A. The force of contact is represented by OA which is opposed by an equal reaction force at B, the resultant being OC. This resultant tends to move the roller towards the wider edge of the wedge. $OC = 2OA \sin \alpha/2$, the value of which is balanced by frictional forces at A and B. These forces F are equal to $F \cos \alpha/2$, which again may be set down as $F \cos \alpha/2 = OA \sin \alpha/2$, so that $F = OA \tan \alpha/2$. If $\beta =$ the friction angle, whose tangent is equal to the coefficient of friction, then the resistance to slip at B is a maximum at $OA \tan \beta$. It will be seen that the wedge angle α must not exceed twice the friction angle; a similar clutch was fitted to the earlier G.M. truck hydraulic transmission.

Considerable attention has been devoted in the past to the development of the hydro-kinetic principles of transmission, more recently the hydrostatic types have become prominent. Analysis of the two principles reveals the fact that whereas in the kinetic type the fluid travels at high speed but at a comparatively steady velocity, in the hydrostatic transmission the operating fluid is subjected to high frequency pulsation, with the result that the consequent frothing and heat generation seriously reduces the mechanical efficiency of the machine. Such a condition does not occur to the same degree in the kinetic type, and therefore the efficiency of that type exceeds that of the hydrostatic transmission. Present developments, however, are tending to reduce this pulsation and when this has been achieved, the main objections to the hydrostatic type will have been eliminated.

Welding Rear Axle Housings

IN present-day Chevrolet cars, the body of the rear-axle housing is made from 0·146in cold-rolled, carbon-steel sheet stock; it is formed in two halves joined longitudinally by submerged-arc welding. Subsequently, a carbon-steel flange must be welded to each end of the housing. This, one of the more critical welds in the manufacture of rear-axle housings, is now made by mechanised, straight-polarity, inert-arc, consumable-electrode welding.

Previously, the particular weld concerned was made first with the open-arc process and later with the submerged-arc process. Both processes were mechanised, but net production per welding machine with open-arc welding was limited to some 45 housings per hr, and other difficulties included porosity with severe undercutting and leakage through the weld. Production was faster when submerged-arc welding was adopted, but here again difficulties were caused by the flux and occasional starting failures. The inert-arc process now employed was selected as best fitting the requirements of high-speed production, simplicity of operation, achieving of a leak-proof weld structurally strong enough to cope with the conditions encountered in fast, modern cars, absence of difficulties, and savings in cost through the realization of 100 per cent production efficiency against the 60 per cent obtained hitherto.

The operator first places the two flanges of forged 1020 carbon steel in retainers which are part of the welding machine. He then positions a housing of 1010–1008 cold-rolled carbon steel in the machine and mechanically presses both flanges over each end of the housing. An automatic mechanism, started by a push-button, moves the housing with flanges into position for welding, which is performed automatically while the operator prepares another housing and flanges. Pushing of the button again causes the welded housing to be ejected from the machine while the next unit moves into the welding position.

Net production per machine has been raised from 60–65 to 100–105 per hr. Handling operations have been reduced to a minimum, welding is automatic, and further simplification results from the ease of inspecting welds visually. There are no complications with flux-handling equipment. Welds are porosity-free, leak-proof with respect to grease, and have an average shear strength of some 75,000 lb; there is a high degree of fusion between weldment and base metal. Difficulties with the motor-generator sets and controls originally employed with the inert-arc process have been eliminated by the substitution of a constant-potential, rectifier type of supply. Initial shielding problems have also been overcome. Rejects are now limited to some 0·5 per cent of total production. Cost savings have been considerable through the elimination of the bottleneck in axle production caused by former welding methods and through the increased overall efficiency.

Industrial Research

General Motors Corporation Technical Centre

WHAT must rank as one of the world's greatest industrial research centres is the Technical Centre of General Motors Corporation at Warren, Michigan, which has recently been completed. It occupies a 330 acre area, and includes 25 buildings, ranging from gatehouses to large laboratories and shop buildings. More than 4,000 engineers, research workers, stylists, designers and other specialists are employed.

At the Technical Centre there are four central staff organizations—research, engineering, styling and process development. The research section is concerned with fundamental information which may prove useful to any of the technical groups within the GM. Its work includes metallurgy, chemistry, physics and many phases of mechanical engineering. The engineering sections concentrate on long-range development work on automobile engines, automatic transmissions, body structures, ordnance vehicles and vehicle components. This group investigates special design projects which are beyond the research stage but not near enough to production to justify divisional development.

The styling department maintains separate studies for each GM automobile division—Chevrolet, Pontiac, Oldsmobile, Buick, Cadillac and GMC Truck and Coach. In the process development section, work is carried out on engineering studies and exploratory projects to improve manufacturing techniques and processes, with a view to improving plant efficiency, and product quality and reducing product costs. In general, the emphasis at the Technical Centre is on advanced engineering and research. Engineering department projects usually look ahead at least three to five years, while research projects extend even further into the future.

The various building groups are ranged three sides of a 22 acre artificial lake. None of the buildings is higher than three storeys. At the north end of the site there are special purpose structures, including a gas turbine research building, a wind tunnel and an isotope laboratory for work with radioactive materials. Along the eastern border of the site there is a one mile check road with loops at the northern and southern ends. It is used for quick tests on vehicle developments, before more comprehensive tests are carried out at the GM proving grounds.

One of the most interesting departments is the hydraulics laboratory of the engineering section, where valuable data is obtained on how to design better automatic transmissions and torque converters for automobiles, trucks and buses. This laboratory has four working levels. In the subbasement there is supply reservoir of 7,500 gallons capacity. The main test area is on the basement level with tanks, channels and flow systems elaborately instrumented for

The new technical centre of the General Motors Corporation at Warren, Michigan

accurate measurement. The main control console is on the ground floor. On the first floor there is a supply tank with a capacity of 3,700 gallons. When a test is being carried out, water from this tank falls straight down 25 ft into the main test area at carefully controlled constant pressure. A flow rate as high as 2,500 gallons per minute can be obtained. Engineers in the main test area can measure this flow of water through a nozzle fitted with various shapes of blades As water flows over the blades, it is possible to determine turbulence, eddies and other flow characteristics, which all serve to indicate whether the blades are designed to give the maximum efficiency in an automobile torque converter.

Another interesting device is a flow table which is in effect a shallow rectangular tank, 8 ft wide, 8 ft long and 4 ft deep, on which water can be watched flowing from one side to the other at a rate of 1,000 gallons per minute. As the water flows at a depth of from 6in to 12in, it courses through a series of "straightening blades" and swirls around oversize wooden turbine blades, which are ten times larger than the blades actually used in a standard automobile torque converter. Dye is injected into the water so that the flow pattern can be seen in colour. By changing the angle of the oversize blades it is possible to determine the comparative efficiencies of various blade shapes.

The flow table for use in developing torque converter blades

Part of the interior in the styling auditorium. This and other buildings have special air conditioning systems developed by Thermotack Ltd. of Glasgow

Another recent development for which the Centre is responsible is the use of zebra-like stripes of light for detecting surface defects in metal bodies. The car bodies on the way to final assembly pass through a tunnel of evenly spaced light bands which almost circle the conveyor line. These alternating light and dark bands "comb" the body, and faults are more readily seen because the flaws alter or interrupt the uniform light and shadow patterns reflected on the metal.

Before the introduction of this development, the bodies were bathed in almost uniform light. Tests, however, showed that too much light actually tended to hide the flaws and also resulted in eye fatigue. It was discovered that the greater the angle of reflection, the more noticeable the flaws became, and it was found that defects could be best seen when the edge of a beam of light fell on them. It was this latter discovery that prompted the use of alternating light and dark bands of light, which, in effect, provide a continuous group of light edges.

Air conditioning

It is a matter of great interest that a British company, Thermotank Ltd., Helen St., Glasgow, W.1, were called to co-operate in the development of a highly efficient system of air-conditioning. The substantial saving in space that was made possible by the Thermotank high-velocity system led to a considerable economy in the final cost. Not only were the smaller ducts that are used in this system easier and cheaper to install, but in addition a high velocity system obviates any need for the services required by local cooling units.

After careful research, a high-velocity, double-duct air conditioning system was evolved by Thermotank Ltd. It uses their improved techniques of balanced pressure controls, noiseless mixing boxes, streamlined ducts and high-velocity ejection outlets. In principle, the system is based

The entrance to the research building

on one hot and two cold vertical ducts concentrated in the spine of the building, and rising to horizontal take-off ducts, which traverse the buildings along the corridor ceilings. Mixing boxes are situated at intervals above the corridors, and are connected to the high-velocity room outlets by flexible hoses. The installations are designed to supply conditioned air at 5,000 ft per minute from a central cooling plant. They allow air changes of between six and fifteen times per hour according to the function and aspect of the building with the use of not less than 20 per cent of outside air.

Individual room temperature control is generally available throughout the scheme to provide for changes in use and to maintain comfort conditions all the year round. Relative humidity control is also provided. During the summer, internal temperatures will normally be maintained at 80 deg F dry bulb temperature, 67 deg F wet bulb (30 per cent relative humidity) when outside temperatures are at a maximum of 95 deg F dry bulb and 75 deg F wet bulb. During the winter with outdoor conditions at a minimum of minus 10 deg F dry bulb, the internal temperatures will be at 70 deg F dry bulb and 53 deg F wet bulb (30 per cent relative humidity). The entire system is automatically controlled, and uses outside air direct, when temperatures allow, in order to save power consumption. A time switch can be set so that at nights or week-ends the plant will operate at half volume.

Typical of the Thermotank installations used in the scheme is that fitted in the Research Administration building. This plant is capable of handling 60,000 c.f.m. of air, of which there is a maximum of 40,000 c.f.m. re-circulated air and 20,000 c.f.m. outside atmosphere. The latter quantity represents an average outdoor air rate throughout the two floors treated in this particular building to give some three changes of air per hour, or 80 c.f.m. per person for a maximum of 250 persons in the building at any one time. The refrigerated water cooling plant has a capacity of 188 tons. It cools 660 gallons of water per minute. Under maximum load conditions, the steam consumption is 3,280 lb per hour at 10 lb/in^2 pressure, although when boosting from cold, the equipment's consumption rises to 6,000 lb of steam per hour. A heating and ventilation system is installed in the basement of the building. It consumes 1,000 lb of steam per hour.

PRECISION-FORGED GEARS

A German Development

A PROCESS has recently been developed in Germany for the production of precision-forged gears that can be used for automobile differentials. Bevel gears with straight or spiral teeth, hypoid gears, elliptical gears, compound spur and bevel gears, and gear parts incorporating notches and cam profiles can all be produced.

To begin with, the gear blank is cut from round stock and machined to a predetermined volume. This operation, of course, also removes surface oxides and scale. The machined blank is heated to about 3,000 deg F in an inert atmosphere to prevent scale formation. The heated blank is then squeezed into the die cavity on a mechanical press. After forging, the work is cooled and slightly tempered.

Five German steels have been used with success for forged gears. These gears have an improved structure which results in teeth that are harder and stronger than cut teeth. To give the required degree of precision, the forging die is mechanically and chemically copied from a hardened and ground master gear of dimensions that have been modified to allow for shrinkage during forging and heat treatment. Despite inaccuracies in the profile of the forged tooth because of the impossibility of making exact compensation for shrinkage, tooth accuracy can be held to ± 0.002in. This is acceptable for gears operating at speeds up to 1,200 r.p.m.

Experience has shown that precision forging is less costly than machining gears, providing the cost of machining the teeth is high in proportion to the total cost. The forging process is not generally economical for the production of less than 500 parts. On spiral gears the teeth must be lapped after forging.

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CALCULATION OF THERMODYNAMIC CHARTS

A New Method of Solving Equilibrium Equations Assists Evaluation of Ideal Efficiencies

K. WEISS

THERMODYNAMIC charts such as developed by Hershey, Eberhardt and Hottel are of the greatest help when the problem of determining the ideal efficiency of an internal combustion engine arises. Unfortunately, these charts are constructed for three mixture-strengths of octane and air and certain octane mixtures only. If fuels other than octane are used, or special air-fuel ratios are required, new charts have to be calculated, based on the thermal data available.

A. S. Leah* has shown how to construct the necessary lines to find the ideal efficiency and mean effective pressure for an engine, following the Otto cycle using ethyl alcohol as fuel. To plot graphs of internal-energy and heat released during reaction against temperature, for the products of combustion, it would be necessary to solve the equilibrium equations in the neighbourhood of the maximum temperature of the cycle. This can be done by using the method of mathematical iteration, which is a slow method when the number of unknowns is greater than four, as in the case of total dissociation. A new tabular method will be described, which, in the author's opinion, takes less time than the methods used so far.

Calculation of the dissociation quantities

To calculate the dissociation quantities at a certain temperature, the following dissociation equations have to be solved:

$$\frac{[CO]^2 \times [O_2]}{[CO_2]^2} = K_1 \quad .. \quad .. \quad .. \quad .. \quad (1)$$

$$\frac{[CO] \times [H_2O]}{[CO_2] \times [H_2]} = K_2 \quad .. \quad .. \quad .. \quad .. \quad (2)$$

$$\frac{[OH]^2 \times [H_2]}{[H_2O]^2} = K_3 \quad .. \quad .. \quad .. \quad .. \quad (3)$$

$$\frac{[H]^2}{[H_2]} = K_4 \quad .. \quad .. \quad .. \quad .. \quad .. \quad .. \quad (4)$$

$$\frac{[O]^2}{[O_2]} = K_5 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (5)$$

$$\frac{[N]^2}{[N_2]} = K_6 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (6)$$

$$\frac{[NO]^2}{[O_2][N_2]} = K_7 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (7)$$

The quantities in brackets are the partial pressures of the constituents, and K_1 to K_7 the equilibrium constants for an assumed temperature, "T".

If combustion takes place at constant volume, as shown in Fig. 1, the maximum pressure "p" for an absolute temperature "T" may be found as follows:

$$144\ p\ V_2 = m_p\ GT \quad .. \quad .. \quad .. \quad .. \quad (8)$$

$$144\ p_1 V_1 = m_m\ GT_1 \quad .. \quad .. \quad .. \quad .. \quad (9)$$

therefore:

$$p = p_1 \times \frac{V_1}{V_2} \times \frac{T}{T_1} \times \frac{m_p}{m_m} = p_1 \times r \times \frac{T}{T_1} \times \alpha \quad (10)$$

In the above equations "p" and "T" represent the pressure and temperature of the products of combustion, without regard to the true final value p_2 and T_2, "r" the compression ratio and "α" the molecular ratio. This is the ratio of the number of molecules of the products of combustion to the number of the molecules of the mixture before combustion.

If "m_p" is replaced by the number of mols of a constituent, then "p" in equation (10), becomes the partial pressure of it and its value can be inserted in the various equilibrium equations. In cycle calculation for petrol engines it is usual to make the mixture "m_m" one lb-mol, while in Diesel engines the fresh air and the residual gases are kept as one lb-mol. Thus "m_m" before combustion will be higher, corresponding to the amount of fuel injected. Equation (1) can, therefore, be written:

$$\frac{\left[p_1 \times r \times \frac{T}{T_1} \times \frac{m_{co}}{m_m}\right]^2 \times \left[p_1 \times r \times \frac{T}{T_1} \times \frac{m_{o2}}{m_m}\right]}{\left[p_1 \times r \times \frac{T}{T_1} \times \frac{m_{co2}}{m_m}\right]^2} = K_1$$

or

$$\frac{[m_{co}]^2}{[m_{co2}]^2} \times p_1 \times r \times \frac{T}{T_1} \times \frac{m_{o2}}{m_m} = K_1 \quad .. \quad (11)$$

In equation (11) m_{co}, m_{co2}, m_{o2}, are the number of lb-mols of CO, CO_2 and O_2 in the products of combustion

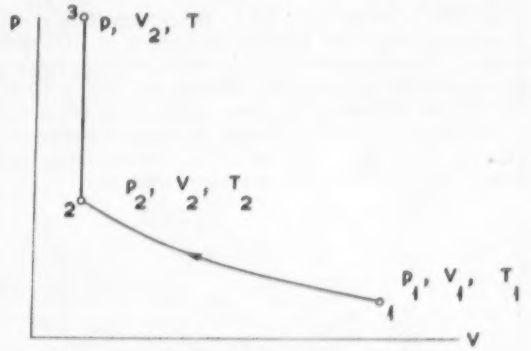


Fig. 1. Curve for combustion and constant volume

at the assumed temperature "T", and "p_1" the inlet pressure in atmospheres. Let x, y, z, 2p, 2q, 2s and t be the number of lb-mols of CO_2, H_2O, OH, H, O, N and NO in the products of combustion and a, b, c, and d represent the number of lb-mols CO_2, O_2 H_2 and N_2 if no dissociation were to occur, then the products of the combustion gas at point 3 in figure 1 will be:

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO + (a - x)CO + \left(b - \frac{x + y + z + t + 2q}{2}\right)O_2 + \left[c - \left(y + \frac{z}{2} + p\right)\right]H_2 + \left(d - \frac{t + 2s}{2}\right)N_2 \quad .. \quad (12)$$

and "m_p" then becomes

$$\left(a + b + c + d - \frac{x}{2} - \frac{y}{2} + p + q + s\right) \text{ lb-mols.}$$

* A. S. Leah. The Ideal Efficiency of Petrol Engines. I.M.E. Proceedings 1949 Vol. 161

The equilibrium equation (11) can now be written as:

$$\left(\frac{a-x}{x}\right)^2 \times \left(b - \frac{x+y+z+t+2q}{2}\right) \times \left(p_1 \times r \times \frac{T}{T_1} \times \frac{1}{m_m}\right) = K_1 \quad .. \quad .. \quad .. \quad (13)$$

The product

$$\left(p_1 \times r \times \frac{T}{T_1} \times \frac{1}{m_m}\right)$$

is a constant as the values p_1, r, T, and m_m are given and should be called C_1.

The equilibrium equations can then be written in their final form:

$$\left(\frac{a-x}{X}\right)^2 \left(b - \frac{x+y+z+t+2q}{2}\right) = \frac{K_1}{C_1} = A \quad (14)$$

$$\left(\frac{a-x}{X}\right) \frac{y}{\left[c - \left(y + \frac{z}{2} + p\right)\right]} = K_2 \quad (15)$$

$$\frac{z^2}{y^2}\left[c - \left(y + \frac{z}{2} + p\right)\right] = \frac{K_3}{C_1} = B \quad (16)$$

$$\frac{4p^2}{\left[c - \left(y + \frac{z}{2} + p\right)\right]} = \frac{K_4}{C_1} = C \quad (17)$$

$$\frac{4q^2}{\left[b - \frac{x+y+z+t+2q}{2}\right]} = \frac{K_5}{C_1} = D \quad (18)$$

$$\frac{4s^2}{d - \frac{t}{2} - s} = \frac{K_6}{C_1} = E \quad (19)$$

$$\frac{t^2}{\left(b - \frac{x+y+z+t+2q}{2}\right)\left(d - \frac{t}{2} - s\right)} = K_7 \quad (20)$$

To solve equation (14)–(20) by the new method, one has to estimate firstly the number of lb-mols of CO_2 in the products of combustion represented as "x" in above formulae. As dissociation increases with increase of "T", a value of "x" may be selected within close limits of the actual value.

Assume for the first calculation the number of lb-mols of CO_2 and insert this value, called "x_1", in the seven equilibrium equations, then they may be written as:

$$\left(b - \frac{x+y+z+t+2q}{2}\right) = A\left(\frac{x_1}{a-x_1}\right)^2 = F \quad (21)$$

$$\frac{y}{\left[c - \left(y + \frac{z}{2} + p\right)\right]} = K_2\left(\frac{x_1}{a-x_1}\right) = G \quad (22)$$

therefore: $\left[c - \left(y + \frac{z}{2} + p\right)\right] = \frac{Y}{G}$ .. .. (23)

but $\left[c - \left(y + \frac{z}{2} + p\right)\right] = B \times \frac{y^2}{z^2}$ (equ. 16) .. (24)

and $\frac{y}{G} = B\frac{y^2}{z^2} \quad \therefore y = \frac{z^2}{BG}$ .. .. (25)

$$4p^2 = C\left[c - \left(y + \frac{z}{2} + p\right)\right] = C\frac{y}{G} = C\frac{z^2}{BG^2} \quad (26)$$

$$2p = \frac{z}{G}\sqrt{\frac{C}{B}} \quad .. \quad .. \quad .. \quad .. \quad .. \quad (27)$$

Equation (23) can now be expressed in terms of "z" as

$$c - \frac{z^2}{BG} - \frac{z}{2} - \frac{z}{2G}\sqrt{\frac{C}{B}} = \frac{y}{G} = \frac{z^2}{BG^2}$$

or $z^2\left(\frac{1}{BG^2} + \frac{1}{BG}\right) + \frac{z}{2}\left(1 + \frac{1}{G}\sqrt{\frac{C}{B}}\right) - c = 0$

$$z^2\frac{(1+G)}{BG^2} + \frac{z}{2}\left(1 + \frac{1}{G}\sqrt{\frac{C}{B}}\right) - c = 0$$

$$z^2 + \left(\frac{BG}{1+G}\right)\left(G + \sqrt{\frac{C}{B}}\right)\frac{z}{2} - cG\left(\frac{B \cdot G}{1+G}\right) = 0$$

if $\frac{BG}{1+G} = H$ .. .. (28)

then: $z^2 + H\left(G + \sqrt{\frac{C}{B}}\right)\frac{z}{2} - cGH = 0$ .. (29)

and $z_{1,2} = -\frac{H\left(G + \sqrt{\frac{C}{B}}\right)}{4} \pm \sqrt{\frac{H^2\left(G + \sqrt{\frac{C}{B}}\right)^2}{16} cGH}$ .. .. (30)

as: $4q^2 = D\ F$, equ. (18) $\therefore 2q = \sqrt{DF}$ .. (31)

Using the positive root value only "y" and "p" can now be found by inserting "z_1" in equations (25) and (27).

From equation (21) "t" can then be found:

$$b - \frac{x_1 + y + z + t + 2q}{2} = F$$

and $t = 2F - 2b - x_1 - y - z - 2q$ .. .. (32)

after finding the value of "t", "s" can be calculated using equations (19)

$$4s^2 = Ed - \frac{E}{2}t - Es \simeq 0$$

$$s_{1,2} = -\frac{E}{2}\left(\frac{1}{4} \pm \sqrt{\frac{1}{16} + \frac{d - \frac{t}{2}}{E}}\right) \quad .. \quad (33)$$

and then $K_7 = \frac{t^2}{F \times \left(d - \frac{t}{2} - s\right)}$ .. .. (34)

In most cases, the value of "s" may be neglected, being too small to show any real heat losses due to dissociation of "N_2". Finally by inserting the values "t" and "s" into equation (34) the value of "K_7" may be found. If it does not correspond with the original value, a new value "x_2" has to be selected and the calculations repeated.

Using a tabular method, one quickly approaches the value of "K_7" and the corresponding values x, y, z, t, 2p, 2q and s should satisfy the seven equilibrium equations.

Example

A port-controlled two-stroke Diesel engine, with blower uses 46% excess air at full load. At the beginning of compression, the charge may consist of 85% fresh air and 15% residual gases; the latter have a ratio of 70% combustion gases and 30% air. All ratios are per weight and the air in the residual gases does not take part in the following combustion process.

Neglecting the small difference in molecular weight between air and the exhaust gases, the combustion equations for this problem using "$C_{12}H_{26}$" as fuel can be written as:

$$0{\cdot}0066C_{12}H_{26} + 0{\cdot}85\ Air + (0{\cdot}0133CO_2 + 0{\cdot}0143H_2O + 0{\cdot}0094O_2 + 0{\cdot}1130N_2 = 0{\cdot}0925CO_2 + 0{\cdot}1001H_2O + 0{\cdot}0655\ O_2 + 0{\cdot}7830\ N_2 \quad .. \quad .. \quad .. \quad (35)$$

The above quantities are the number of lb-mols of CO_2, H_2O, O_2 and N_2 if no dissociation were to take place.

To find the dissociation quantities at 5000°R, the following data are used:

The compression ratio r = 14:1 (this is ratio of the remaining total volume after the piston has just covered the exhaust port to the clearance volume); the inlet temperature T_1 = 700°R, which is the temperature of fresh air, mixed with the residual gases, the inlet pressure p_1 = 1·155 atmospheres, and the dissociation coefficients at 5000 deg R as follows:

Values of "K" at 5000 deg R.

$K_1 = 1{\cdot}995 \times 10^{-2}$
$K_2 = 6{\cdot}546$
$K_3 = 1{\cdot}738 \times 10^{-2}$
$K_4 = 5{\cdot}888 \times 10^{-2}$
$K_5 = 2{\cdot}692 \times 10^{-7}$
$K_6 = 2{\cdot}089 \times 10^{-3}$
$K_7 = 8{\cdot}710 \times 10^{-8}$

Equation (12) can be written as:

$$xCO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO + (0{\cdot}0925 - x)CO + \left(0{\cdot}0655 + 0{\cdot}04625 + 0{\cdot}05005 - \frac{x + y + z + 2q + t}{2}\right)O_2 + \left[0{\cdot}1001 - \left(y + \frac{z}{2} + p\right)\right]H_2 + \left(0{\cdot}7830 - \frac{2s + t}{2}\right)N_2 =$$

$$x\,CO_2 + yH_2O + zOH + 2pH + 2qO + 2sN + tNO + (0{\cdot}0925 - x)\,CO + \left(0{\cdot}1619 - \frac{x + y + z + 2q + t}{2}\right)O_2 + \left[0{\cdot}1001 - \left(y + \frac{z}{2} + p\right)\right]H_2 + \left(0{\cdot}7830 - \frac{2s + t}{2}\right)N_2 \quad .. \quad .. \quad (36)$$

Therefore:

$a = 0{\cdot}0925$
$b = 0{\cdot}1619$
$c = 0{\cdot}1001$
$d = 0{\cdot}7830$

$$A = K_1 \times \frac{T_1}{T} \times \frac{m_m}{p_1} \times \frac{1}{r} = \frac{1{\cdot}955}{10^2} \times \frac{700}{5000} \times \frac{1{\cdot}0066}{1{\cdot}155} \times \frac{1}{14} = 1{\cdot}7387 \times 10^{-4}$$

$$B = \frac{K_2}{C_1} = 1{\cdot}5207 \times 10^{-5}$$

$$C = \frac{K_4}{C_1} = 5{\cdot}152 \times 10^{-6}$$

$$D = \frac{K_5}{C_1} = 2{\cdot}3555 \times 10^{-6}$$

$$E = \frac{K_6}{C_1} = 1{\cdot}8278 \times 10^{-9}$$

Assume for the first calculation a value of "x_1" then "K_7" can be calculated after finding the corresponding values of y, z, t, and 2q only as "s" in most cases can be neglected.

The accompanying table gives values of "K_7" as derived after selecting various values of "X_1".

Conclusion

The tabular method gives a fairly quick way of finding the dissociation quantities, especially if a calculating machine is available. The values of a, b, c and d are constant for any temperature "T" and the remaining constants A, B, C, D and E remain unchanged for each temperature value of "T".

If the dissociation values for different temperatures "T" have to be calculated, it has to be remembered that with increase of "T" the quantity of "CO" increases therefore "x" has to be decreased and a new table can then be calculated. For most problems it is sufficient to stop calculations, if a value of "K_7" has been found which does not differ much from the original figure given in the tabulation at the top of this page.

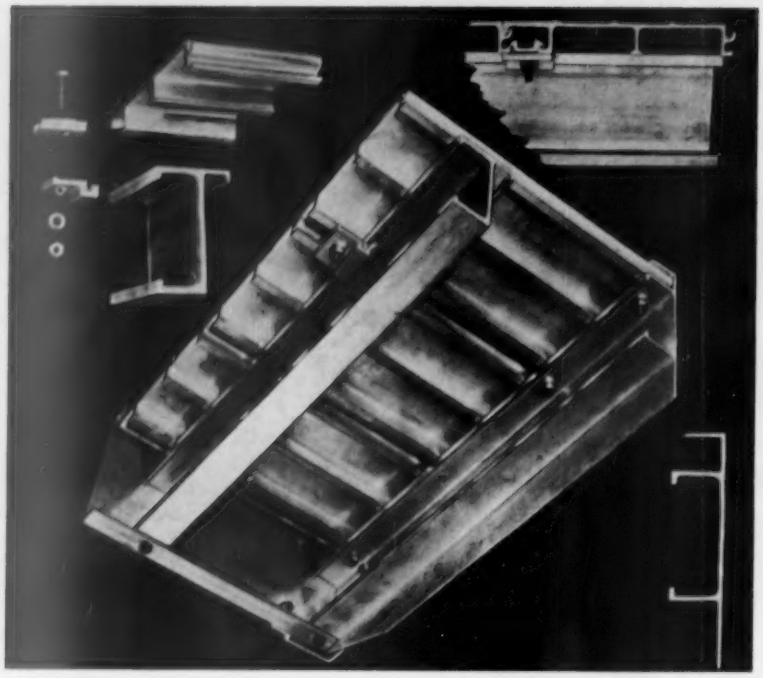
VALUES USED IN FINDING VALUES OF K_7

	1st Trial	2nd Trial	3rd Trial	Final Trial
1. x_1	0·0870	0·0875	0·0876	0·08758
2. $a - x_1$	0·0055	0·0050	0·0049	0·00492
3. $\frac{x_1}{a - x_1}$	15·818	17·5000	17·8775	17·800
4. $\left(\frac{x_1}{a - x_1}\right)^2$	250·2091	306·25	319·605	316·843
5. $F = A \times \left(\frac{x_1}{a - x_1}\right)^2$	$4{\cdot}3326 \times 10^{-3}$	$5{\cdot}3247 \times 10^{-3}$	$5{\cdot}5569 \times 10^{-3}$	$5{\cdot}5089 \times 10^{-3}$
6. $G = K_2\left(\frac{x_1}{a - x_1}\right)$	103·5446	114·555	117·026	116·519
7. $H = \frac{B \cdot G}{1 + G}$	$1{\cdot}5061 \times 10^{-5}$	$1{\cdot}5061 \times 10^{-5}$	$1{\cdot}5078 \times 10^{-5}$	$\overset{A}{1{\cdot}50775} \times 10^{-5}$
8. z as per eq. (30)	0·01209	0·01270	·001285	0·01283
9. $y = \frac{z^2}{B \cdot G}$	0·09289	0·09258	0·09278	0·092899
10. $p = \frac{z}{2 \cdot G}\sqrt{\frac{C}{B}}$	not cal.	n.c.	0·000101	0·0001036
11. $2q = \sqrt{F.D.}$	0·00101	0·00119	0·00114	0·001139
12. $t = 2b - 2F - x - y - z - 2q$	0·0445	0·0236	0·0185	0·019
13. s = as per eq. (33)	n.c.	n.c.	n.c.	0·000019
14. $K_7 = \frac{t_a}{F \times \left(d - \frac{t}{2} - s\right)}$	$6{\cdot}0118 \times 10^{-3}$	$1{\cdot}357 \times 10^{-3}$	$7{\cdot}961 \times 10^{-3}$	$8{\cdot}652 \times 10^{-3}$

Note: The correct value of K^7 at 5000 deg R taken from Table 1 is $8{\cdot}710 \times 10p^5$

Simplified Body Building

Specially Developed Light-alloy Sections for Commercial Vehicles



A NEW series of light-alloy extruded sections is now available for building bodies for commercial road vehicles. They form the basis for a very useful range of kits from which flat-platform and drop side bodies can be built quickly, economically and without the need for skilled labour. The sections are produced by Northern Aluminium Company Ltd. for E. J. Holmes Ltd., 113—115 North Street, Romford Essex, and are obtainable from either of these organisations. They comprise two floor planks, two cross-bearers, two side, raves, floor-holding clips and a drop side framing member. In sum, they present a significant advance in body building techniques, particularly as regards economy and ease of assembly.

The floor sections, which may be flat or ribbed, are designed for fixing either by bolts passing through the top flange of the cross-bearer, or by means of the special floor-holding chip sections. These clips can be slipped into place after the floor planks have been laid; they locate on lips on the cross-bearer sections. Each clip clamps the edges of two adjacent floor planks to make an extremely quick and effective joint. The upper part of the clip will take the head of a standard hexagon bolt and prevent it from turning. A tongue-and-groove arrangement is incorporated in the planks; this helps to exclude dust and moisture. If jointing compound is used, a complete seal can be made.

There are two types of cross-bearers. Those for intermediate positions are of lipped channel section, with an additional flange that makes them almost equivalent to a channel and an angle placed back-to-back, to give in effect a 'J' section. The reason for the additional flange is that the main channel portion is light and the extra flange allows the section to be drilled for floor fixing and still retain sufficient strength. A further advantage of this shape is that it allows two floor-holding clips to be used at each joint, and a much more rigid connection to be obtained. The section for front and rear bearers is similarly based on a lipped channel, but incorporates a floor capping angle and a shroud plate as well. This design leads to a great saving in assembly labour. When used at the front of the vehicle, it also adds considerable stiffening to the body and headboard.

The two rave sections and the framing member are conventional. They are designed to give a neat appearance with the minimum of effort. For example, shallow grooves are provided to give drill-point location wherever it is necessary to fit bolts or rivets. The accompanying illustration shows a short length of lorry body made up from the extruded sections produced by Northern Aluminium Company Ltd. At the top left the illustration shows a short length of the 'J' section cross-bearer, with a floor plank and floor-holding clip; the top right shows the plank profile and its method of attachment; and the bottom right shows the combined end cross-bearer, floor capping angle and shroud plate. All the parts needed to build complete bodies can be ordered cut to length, and a minimum of equipment is needed for their assembly.

Special Numbers

THE Show Review number of the *Automobile Engineer* will be published on Wednesday, November 28th. It will constitute a critical review of the more interesting exhibits and will have numerous illustrations of special features and design characteristics. This special issue can be obtained by order from newsagents throughout the United Kingdom, price 3s. 6d. net. Readers are reminded that it is necessary to make arrangements with a newsagent to ensure that a copy is secured.

Our associated journal *The Autocar* will publish three special issues in connection with the Show. The first will be a Show Guide. It will be published on October 12th at the usual price, 1/-. On October 19th the Show Report will be published, price 2/-. This issue will contain descriptions of new cars, components, caravans and accessories. The Show Review will be published on October 26th at the usual price, 1/-. It will include a full technical survey of design trends and a final appraisal of the principal exhibits.

PETROL INJECTION

A New System Developed by American Bosch

MUCH experimental work on petrol injection systems to replace the carburettor is being carried out in Britain and the U.S.A. For years designers in many countries have adapted diesel fuel equipment for petrol engines, and, although good results have been obtained, the high cost of the equipment has prevented its widespread use. Now the American Bosch Division of the American Bosch Arma Corporation, Springfield, Massachusetts, has recently released details of a new fuel system in which fuel is injected directly into the engine either at a point in the manifold just above each intake valve or directly into each cylinder itself.

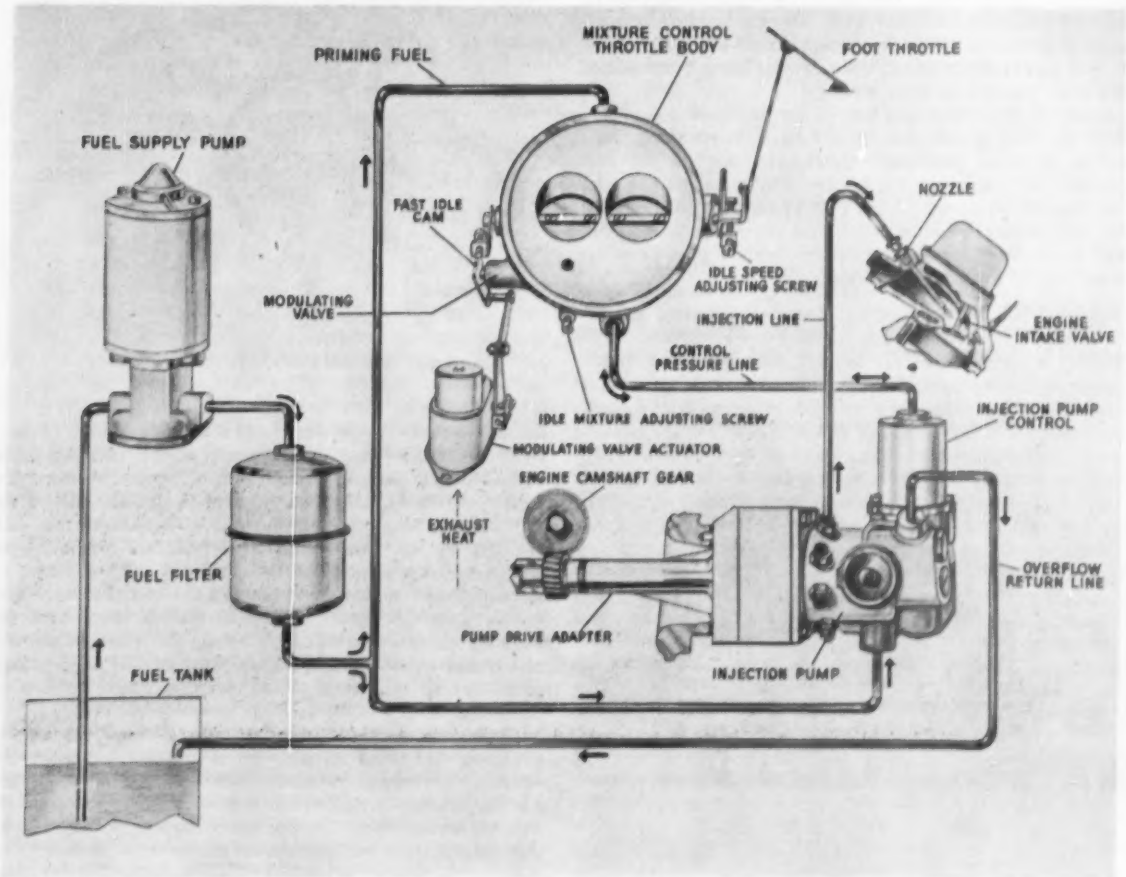
Fuel injection is not a new field for American Bosch. The company carried out considerable research work on the subject before the second world war and during the war helped to develop and produce fuel injection systems for B29 bomber engines. In the post-war period, the company has undertaken a rapidly expanding programme to develop a high-performance system sufficiently inexpensive for automobiles and lorries. The Petroleum Chemicals Division of E.I. Du Pont De Nemours & Company has been testing the new system on a special test car for over a year and reports similar advantages to those claimed by American Bosch.

The latter company realized that the cost of the system had to be drastically reduced if it were ever to gain popularity in the automobile field. The new system consists of an exceptionally small and simplified fuel metering and injection pump, a mixture of fuel-air-ratio control, an electrically driven fuel supply pump, injection discharge lines and nozzles, and a fuel filter.

A diagrammatic view of the system is shown in Fig. 1. The motor-driven supply pump supplies fuel through the filter to the injection pump and maintains fuel pressures at well over the service vapour pressure of petrol. The filter traps any harmful particles in the fuel down to 2 microns in size; it has a service life of approximately 30,000 miles of normal car operation.

According to the company, the system has many potential advantages. More accurate fuel distribution to each engine cylinder results in improved economy, and this also means there is much less dilution to the engine oil with consequently less wear. Furthermore, with certain controls incorporated

Fig. 1. Diagram of the American Bosch petrol injection system



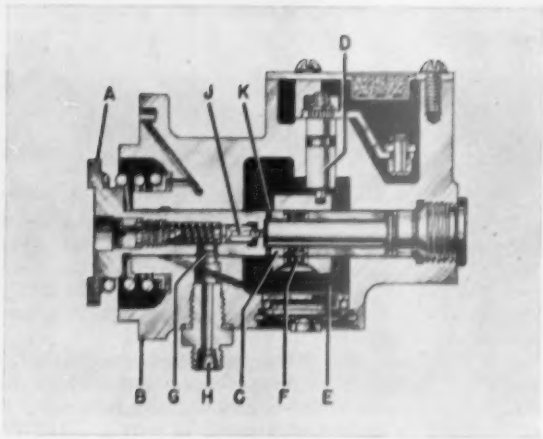


Fig. 2. Sectional view of injection pump

in the American Bosch system, an idling or decelerating engine runs with a much less rich mixture.

With this new system, it is claimed that the available peak horse-power of a standard engine is increased approximately 10 per cent. In addition, a corresponding increase occurs in the torque at lower speeds. The engine speed at which peak torque is developed is lowered about 500 r.p.m. and the torque curve flattens. Acceleration is improved because of more instant response to fast or slow throttle movements. Because it is a pressurized system, less expensive high vapour pressure fuels may be used, yet there is greater freedom from icing and vapour lock. One advantage that will appeal to automobile designers is that with this new system the head room required in the engine compartment can be reduced by 4 to 6 in or more, which will allow for lower bonnet lines and other changes in body styling.

It should be emphasized that all petrol injection development work at present has been done with standard automobile engines. American Bosch has not as yet had available for testing an engine specifically designed to take the greatest advantage of its new system. It is expected that with such a specifically designed engine, at least 25 per cent more output horsepower can be obtained plus a marked improvement in economy of operation.

Injection pump

The sectional view (Fig. 2) of the injection pump proper shows (A) the single unit face-cam and plunger which is reciprocated and rotated by the engine's drive mechanism at one half engine speed. A fuel metering sleeve (C) fitted to the plunger is adjusted by an eccentic shaft (D) linked to the pump control.

In operation, supply fuel enters the sump (E) and during the suction stroke fills the internal volume of the plunger through radial ports (F) in the plunger and sleeve. As the plunger turns, the fill ports are covered at the end of the suction stroke and distributing port (G) registers with one of the outlets (H). At the start of the pumping stroke, fuel trapped in the internal volume of the plunger is forced out through spill ports (K) to the sump for that part of the stroke determined by the position of the metering sleeve. As the plunger lifts further, the spill ports are covered by the sleeve and fuel is then forced through delivery valve (J) and out through a distributing slot (G) to a nozzle line. Injection continues until the end of the stroke and the cycle is repeated for an adjacent outlet.

Injection pump control

The control unit (Fig. 3), which is mounted directly on the pump, receives control pressure from the mixture control

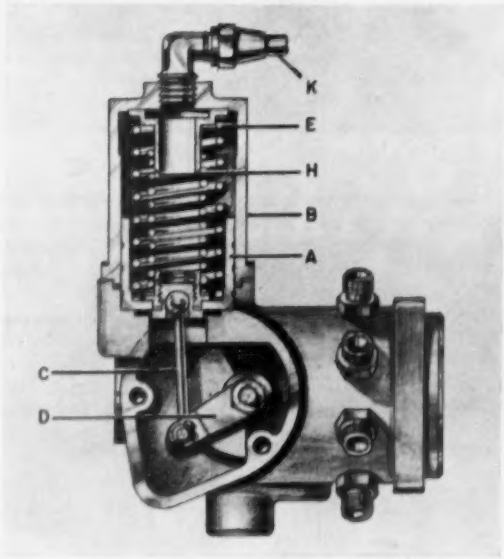
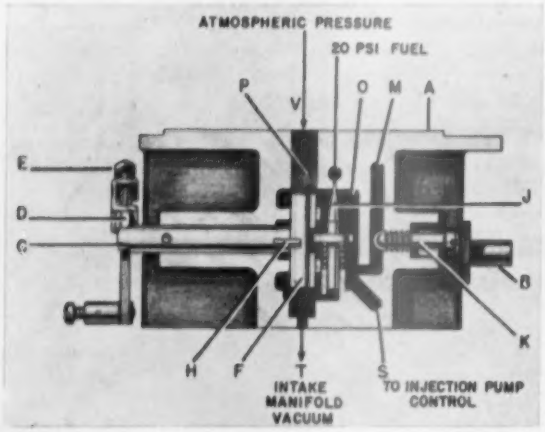


Fig. 3. Sectional view of injection pump control

Fig. 4. Sectional view of mixture control



through a tube (K) and translates it through a link (C) and lever (D) into an angular movement at the metering shaft of the injection pump. At high control pressure corresponding to low manifold vacuum (wide open throttle) the piston (A) is moved down in the cylinder (B) toward maximum fuel position by an outer spring (E). As the engine speed increases, the control pressure is reduced and the piston is forced upward against the spring (E) at a constant rate until the free inner spring (H) reaches its seat, at which time the latter spring adds its effect to spring (E) and the rate of piston travel is reduced. The quantity of fuel injected per stroke is thus a function of the control pressure applied to the piston.

During deceleration, full or partial cut-off of fuel flow to eliminate unburned exhaust gases can be automatically effected by several methods. One method employs an auxiliary piston carried within control piston (A) and biased by a low-rate spring. During normal operation from idle to full throttle, this auxiliary piston moves with piston (A) as a unit. However, when deceleration occurs, the piston

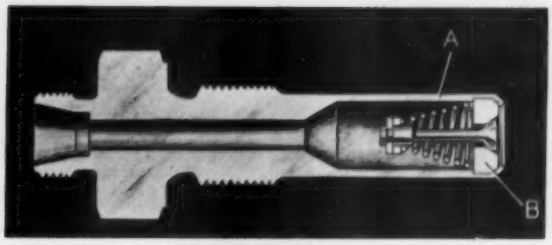


Fig. 5. Sectional view of spray nozzle

moves upward within piston (A) and the fuel pump metering lever is moved to the fuel cut-off position.

Mixture control

The mixture control in its basic and simplest form is merely a manifold pressure control and throttle-body with a chamber feeding a modified manifold pressure to the injection pump control during cold starting and idling. In the sectional view (Fig. 4) of the mixture control, the throttle plates mounted on shaft (B) are not shown in order to portray more clearly the other parts of the unit. The throttle plates are of conventional design with an idling port just above the closed throttle position and connected to passage (M) which enters the control chamber (O) formed in the throttle-body casting. This chamber connects at (S) with the injection pump control.

Idle air flow is adjusted in the usual manner by a screw (E). The idle mixture is adjusted by screw (K) which at idle receives inlet air pressure through idle slot (M) and modifies the control chamber pressure. Above the idle setting, the throttle plate edge rises above the idle port; the idle port becomes subject to manifold pressure and screw (K) has no effect on the engine mixture.

Since a cold engine requires excess fuel during the warming up period, the control pressure must be raised to increase pump delivery. This is accomplished by adjusting a valve (F) which operates within a passage between upper (atmospheric) and lower (intake manifold) pressures. This valve, connected through pin (H) and eccentric shaft (C) to a lever which also carries a fast-idle cam (D) is actuated by a modulating valve actuator controlled by exhaust heat. During normal engine operation the valve (F) is seated at position (P). During cold engine operation, valve (F) is in the position shown (between V and T) which modifies manifold pressure to pump control and the injection pump delivers additional warm-up fuel to the engine until the valve (F) reached position (P).

To ensure good cold-starting, a fuel flow considerably greater than the quantity required for full-load running is needed. This is accomplished by utilizing the supply fuel available at a pressure of 20–30 psi as a primer. A spring-loaded priming valve (J) is mounted adjacent to warm-up valve (F) and is held normally closed on its seat against fuel supply pressure by a spring. An extension on the valve (J) engages in a slot in the valve (F) so that in the cold position a slight additional movement of the shaft (C) allows the valve (F) to engage the priming valve (J) and open it. Fuel then is discharged into the control chamber and enters the intake manifold through passage (T). The additional movement of the shaft (C) necessary to actuate the primer valve (J) may be furnished conveniently by a small solenoid acting through the heat-responsive element and energized by the electric starter circuit. By making the primer solenoid act through the heat-responsive element, priming is prevented when restarting the hot engine.

The spray nozzle (Fig. 5) used is a simple outwardly opening poppet type and consists of a valve assembly (B) fitted into a holder (A) which is available in several lengths to suit various engine and manifold requirements.

WEAR TESTS FOR CYLINDER LINER MATERIALS

IN *A.T.Z.* for April 1956, P. A. Heller and W. Glaser describe a method whereby test material and reference material are investigated simultaneously in the same two-cylinder diesel engine, one cylinder being made of the test material and the other of the reference material. Many of the differences which otherwise arise in the abrasion loading of the two materials are thereby excluded. High circulating speed of the coolant and equal fuel admission produce the same thermal loading, and the number of loading cycles and total sliding distance of pistons and rings can be kept absolutely constant in this manner, the difficulties usually encountered in controlling the numerous variables in normal road or laboratory tests of cylinder liner wear are avoided.

The test unit was a two-cylinder, water-cooled, Deutz diesel engine of 2·2 litre capacity, having an output of 25 h.p. at 1,550 r.p.m., and fitted with wet cylinder liners relatively simple to change. Commercial-quality piston rings of average hardness of 248–262 Brinell were carefully chosen to be closely similar in size, clearance, and tangential stress. Heat load, coolant temperature, oil-sump temperature, output, r.p.m. and fuel consumption were carefully checked. The same fuel and oil were used throughout, fresh oil being put in before each test. In long (100 hr) tests, coolant-outlet temperature was maintained at 60–61 deg C. Prolonged tests may give quite different results from those obtained in frequently interrupted test runs, and some materials were therefore tested both in runs of 100 hr and in intermittent running. Testing conditions are tabulated for both types of test.

Wear measurements were carefully carried out by means of inside calipers accurate to $\pm$ 0·001 mm. Cylinder diameters were measured in four different directions relative to the crankshaft axis and at nine different levels, the average of the four diameter measurements being taken to eliminate cylinder distortion effects, and a special device ensuring that measurements were made always at the same places. After the completion of 100-hr tests, residual internal stresses were very slow to cancel each other out, and final wear figures were obtained by complete measurements on both cylinders daily until there was no further variation from day to day. From these final figures, and the specific weight of the cylinder material, total wear was calculated on a weight basis, as is explained by an example.

Eight different, centrifugally-cast, non-heat-treated cylinder materials were tested, and the results are discussed with reference to basic structure and graphite structure. It is found that wear decreases with increasing hardness, and this relationship is largely independent of basic structure and graphite structure. Austenitic materials with high corrosion resistance are an exception to this rule. A caution is given against attempts to apply the finding indiscriminately; experience with engines indicates, for example, that, with heat-treated or flame-hardened liners, the relationship between wear and hardness does not exist.

MACHINING TITANIUM

Notes on the Techniques to be Employed

ALTHOUGH titanium is not of immediate practical interest to the automobile engineer, there are automobile applications for which its properties are probably better than those of the materials now being used. It is, however, a very costly metal and its adoption for automobile purposes will be delayed until the price is greatly reduced. The production of metallic titanium in this country has been pioneered by Imperial Chemical Industries Limited, and the information contained in these notes is based upon work carried out by ICI.

Machining

Cutting speeds in general should be low and feeds should be as coarse as possible, depending of course upon the strength and rigidity of the tool and the standard of surface finish that is required. Light finishing cuts, particularly if less than 0·02in deep, should be avoided, since the rate of tool wear can be excessive under such conditions. Feeds, particularly in turning, should be coarse by normal standards, and feeds of less than 0·010in per revolution are undesirable. On large diameter material coarse feeds in excess of 0·60in per revolution have been employed successfully with high-speed steel tools. Heavy feeds have, in fact, many advantages, particularly in machining rough cast or forged material, because the material being removed is sheared in front of the cutting edge and the rate of tool wear is significantly reduced.

High-speed steel, cast alloy or tungsten carbide tools may be used. The "super" grades of high-speed steel give goods results in turning work over 6in diameter, particularly when the surface is rough or the cut intermittent. In general, the use of tungsten carbide tools is confined to smaller diameter work on which lighter, more continuous cuts are made, but it may be necessary to use them on larger diameter work in the harder alloys. Frequent regrinding is essential for carbide tools.

Successful results have been obtained with very different tool shapes. Plan shape is relatively unimportant. For tungsten carbide tools, the top rake should be from 6 deg positive to 7 deg negative, depending upon the severity of the operation. Experience has shown that cast alloy tools operate best with 5 deg positive rake and high-speed steel tools with up to 15 deg positive rake. In all cases a relief angle of approximately 7 deg is suggested.

The chief problem in milling arises from chips welding on to each tooth and being carried round with the cutter. This can cause cutter chipping and breakage. To minimise this trouble, it is advisable to employ climb milling whenever possible. This, of course, demands absolute rigidity of work and tool to avoid chatter, but the chip is attached to the cutter tooth by only a thin sliver and is easily broken off.

Titanium may be drilled with high-speed drills. A short drill should be used and the holes should be as shallow as possible. For sizes below ¼in diameter, the best results are obtained with a 140 deg point, but on larger sizes either a 90 deg point or a double-angle point will give better results. In drilling holes of depth exceeding five diameters it is advisable to retract the drill at intervals and clear the swarf. A continuous feed of about 0·002-0·005in per revolution for small diameter holes and of 0·005-0·009in for larger holes should be maintained. Flood lubrication with a heavily chlorinated cutting oil assists in reducing frictional troubles.

Originally, grinding presented some difficulties, but satisfactory techniques have now been evolved. A reduction in wheel speed to one-half or one-third of the conventional speed, in conjunction with the use of a suitable coolant, will usually result in an acceptable grinding ratio. Water-base soluble oils can be used but result in short wheel life. Some chlorinated or sulphurized grinding oils have proved satisfactory, while vapour-phase rust inhibitors of the nitrite-amine type give equally good results.

Vitrified bond A60M wheels have been used at a speed of 1,500 surface feet per minute, and grinding ratios of 10 or more have been obtained while removing as much as 0·08in³ of metal per minute. Abrasive cut-off can be a simple method of parting small diameter rod and bar, provided the work is flooded with coolant, the wheel speed is restricted to a surface speed of 3,000-4,000ft per minute, and the work is rotated to reduce the arc of contact with the wheel. There may however, be practical difficulties in changing machines to low-speed grinding. At conventional speeds wheel wear will inevitably be greater and the surface finish poorer.

Hot working

Titanium and its alloys can be readily hot worked at temperatures generally somewhat lower than those used for steels. They differ from steel however, in that they should be held at high temperatures for only a short time, in order to minimise surface contamination. At temperatures of 700 deg C and less the rate of contamination is relatively low, but it increases rapidly with increase in temperature.

All forging furnace atmospheres contain free or combined oxygen, and some absorption of this element inevitably occurs. In addition to causing visible scaling, some diffusion of oxygen takes place and there is a consequent hardening of a relatively shallow underlying layer. Subsequent operations such as machining will remove this surface layer, and the final product will be of a similar hardness to the original forging stock. Contamination by hydrogen is more serious, since hydrogen has a more rapid diffusion rate than oxygen and generally penetrates across the full section of the work. When this occurs the material can be recovered only by prolonged vacuum annealing. Hydrogen is absorbed from both reducing and oxidizing gas and oil fired furnaces, but at a tolerably slow rate under strongly oxidizing conditions.

Pre-heating time must be kept as short as possible consistent with uniform heating. At the same time, maintaining the material at high temperatures must be avoided. This, of course, entails careful planning of operations and it is preferable to have forging equipment waiting for work than to permit soaking at elevated temperatures. The order of preference for pre-heating atmospheres is : dried air (electric heating), undried air (electric heating), oxidizing oil or gas-fired furnaces, reducing oil or gas-fired furnaces. It is essential to avoid direct flame impingement.

Techniques employed for press and hammer forging of titanium are essentially the same as those for low-alloy steels. As it is highly desirable that the number of reheats necessary should be as few as possible, in order to reduce contamination during the forging operations, good handling methods and plant layout are essential. Repeated light blows or attempts to continue forging at lower temperature should be avoided, since they may promote internal cracking. Trimming should be carried out hot and because each operation must be performed with the least possible pre-heating, the furnace, drop hammer, and trimming press should be stationed as close together as possible.

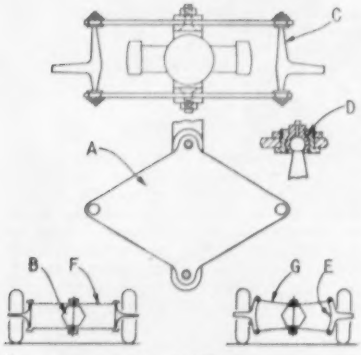
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Wheel suspension and springing

This transverse springing arrangement aims to provide a simple construction having few component parts. Pairs of single-plate springs are used and guiding levers or links articulated to the vehicle framing are not required. Upper and lower plate springs A, of rhomboidal shape, are secured by clamp plates and two common through bolts to the tubular chassis "backbone" member B. In the example shown having a horizontally-opposed engine disposed between the springs, the end of the chassis member is bifurcated and the springs are individually attached.

The springs are free from slots, punchings or perforations except for the attachment holes in lugs outside the clamping area and at the extremities. Front wheel carriers C are of the ball-end type, seating in sockets D mounted in the spring ends.



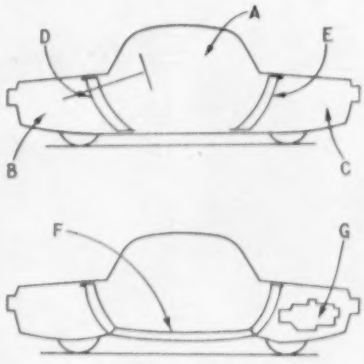
No. 746729

Rear wheel carriers E make connection by rolled spring eyes and pivot pins. Preferably the front springs are arranged as a parallelogram, as at F, while the rear springs are disposed so as to diverge outwardly from a horizontal plane of symmetry containing the wheel axes, as at G. This latter arrangement determines wheel movement in the manner of a swinging axle. *Patent No.* 746729. *Daimler-Benz A. G.* (*Germany*).

Pontoon type body construction

Essential features of this invention are that the transverse walls partitioning the passenger space from the two end portions of the body are spherically curved and that the said walls, extending from the floor plate to the wasitline, are inclined towards their respective neighbouring ends of the vehicle. This construction is claimed to provide great stiffness and stability in the structure and to be completely effective in the suppression of rumbling noises.

Two vehicles are shown schematically. In the first passenger space A, front end B, and rear end C are partitioned by transverse walls D and E. Each wall is spherically curved, with its concavity facing the passenger space, and is inclined from the floor towards its respective end of the vehicle. The second example is of an advantageous construction in which the floor plate F is also spherically curved and



No. 746584

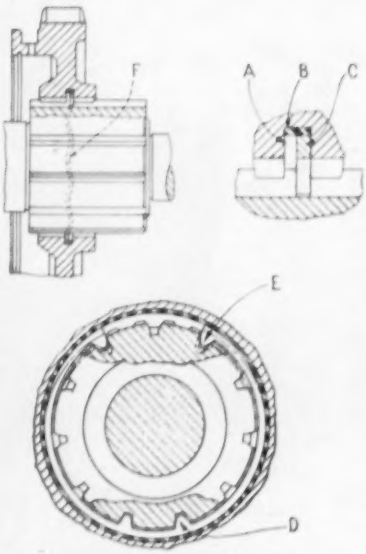
positioned with its concavity facing the passenger space. It shows a rear-mounted engine at G. If desired one or both of the walls D and E may, without departing from the invention, be arranged with the convexity facing the passenger space. *Patent No.* 746584. *Daimler-Benz A.G.* (*Germany*).

Anti-rattle device for sliding gears

Transmission gears slidable on splined shafts must necessarily be provided with reasonable tolerances and consequently are liable to oscillate and rattle when running idly under no-load conditions. To obviate such tendencies a resilient metal ring is arranged to grip simultaneously both gear and shaft. In the example shown the main components are arranged for driving engagement on alternate splines but this is not an essential feature of the invention.

An internal circumferential groove A is cut in the mid-portion of the gear hub. In this groove is an oil-resistant rubber ring B against which is disposed a metal ring C. Internally this ring has a plurality of lugs D and terminal lugs E which on assembly are engaged in the shaft splines The outline of the ring in its unstressed condition is indicated in broken line. To grip the gear, the ring has a wavy configuration, as at F, giving an overall width, when unstressed, slightly greater than the width of the groove.

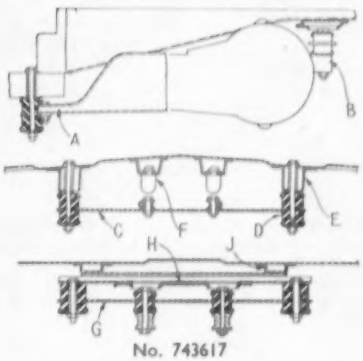
No modification of the normal splines of either shaft or gear is necessary and if the gear is removed the resilient ring will be retained ready for subsequent reassembly. *Patent No.* 746228. *General Motors Corporation* (*U.S.A.*).



No. 746228

Suspension of differential casing

If the differential casing of a De Dion type driving axle is mounted directly on the vehicle chassis or combined chassis and body structure, vibrations and road shocks may be transmitted to the body and constitute a source of noise and discomfort. The use of resiliently mounted suspension bolts will attenuate but not overcome this drawback. As a solution it is arranged to resiliently suspend the casing from two resilient cross members which are also resiliently mounted on the chassis or chassis and body structure.



No. 743617

The casing is provided with two fixing lugs A at the front and two lugs B at the rear. These lugs are not positioned horizontally but lie in a plane passing through the centre of the universal joint between the propeller shaft and the differential gear. At the forward end, lugs A are bolted to a spring blade cross member C, the ends of which are supported in pads D of resilient material fitted with metal caps to receive fixing bolts clamping it to top-hat section ribs E on the body structure. Stops F of a medium-hard plastics material are provided above the casing attachment bolts to dampen and limit rebound.

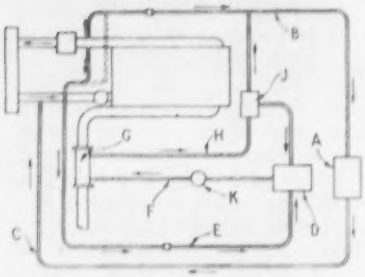
At the rear, lugs B are resiliently attached to spring blade G which is resiliently mounted on a pressed cross member H bolted, riveted, or welded to ribs J on the body.

The arrangement is also applicable to front wheel drive units. *Patent No.* 743617. *Regie Nationale Des Usines Renault* (*France*).

Booster heating system

The characteristic feature of this vehicle heating system is that the auxiliary booster system, utilizing heat transferred from the exhaust gases, is under the control of a thermostatic valve which permits the liquid to be returned to the main system only after it has attained a predetermined temperature. In the diagram, the main heating circuit includes one or more heaters A with a flow pipe B, leading from the engine cooling system return pipe, and a return pipe C to the flow pipe of the cooling system.

Tank D of the booster system is connected by pipe E to the cooling system return pipe and by pipe F to one end of a cylindrical jacket G surrounding the engine exhaust pipe. From the other end of the jacket a pipe H leads to the thermostatic valve J where the fluid is directed either to the main flow pipe B or to the tank D. Pump K circulates fluid from tank D

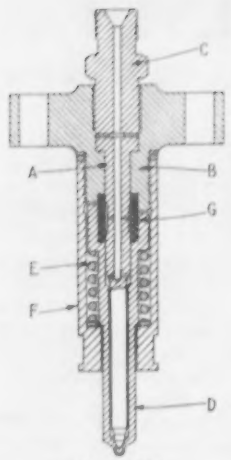


No. 747024

through jacket G and returns it to the tank until the temperature of the fluid leaving the jacket reaches a predetermined value; a non-return valve in pipe E prevents flow back to the cooling system. When the required temperature has been attained, valve J diverts the fluid to the main circuit flow pipe B. A second non-return valve prevents its passage back to the cooling system.

Tank D is provided to ensure an adequate supply of heated fluid is available from the booster system. Any fluctuations in the temperature of the fluid entering the booster system are absorbed and the temperature of the fluid entering the jacket G remains fairly even.

As an alternative to feeding the booster system from the cooling system return pipe, the intake may be from either the upper part of the engine cooling jacket or from the flow pipe B of the main heating circuit, as shown in broken line. *Patent No.* 747024. *Clayton Dewandre Co. Ltd.*



No. 746785

There is no leakage of fuel past the needle valve and consequently no drain hole in the body and no drain pipe are required. The nozzle is self-aligning on the needle valve. The possibility of a secondary injection due to spring oscillation and combustion pressure is eliminated.

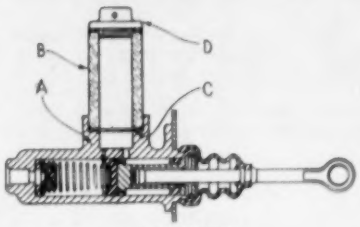
The needle valve A is entered with a tight push fit in the valve body B and clamped against a shoulder by the fuel pipe coupling C. Nozzle D is slideable on the major diameter of the needle valve, projecting from body B, with adequate clearance to permit the nozzle seat to align with the conical face of the needle. At its inner end the nozzle has a portion of larger diameter which provides a shouldered abutment for the spring E, which is retained by sleeve F screwed on the valve body.

Both the end of the nozzle and the valve body are recessed to accommodate a resilient sealing ring G. The needle valve is relieved in way of this ring and radial holes communicate with the bore. Thus the inner surface of the ring is under fuel pressure at each injection impulse to deform the ring by compressing it against all confining surfaces and any leakage of fuel is obviated. *Patent No.* 746785. *Simms Motor Units Ltd.*

Hydraulic brake master cylinder

This master cylinder is intended for mounting relatively high on the dash panel in the engine compartment and for direct operation by a depending pedal in the passenger compartment. In that position it is protected, accessible, and readily visible when the bonnet lid is raised. The fluid reservoir is arranged in unit with the master cylinder, being formed in part by an integral cylindrical socket A and part by a cylinder B of a transparent or translucent plastics material. Cylinder B is screwed into the socket, making a fluid-tight joint by means of a resilient sealing ring C, and is closed by a suitably vented screwed cap D.

Since cylinder B is semi-transparent and is easily visible the amount of fluid in the reservoir can be ascertained by a glance. The metal socket portion of the reservoir provides a safety feature as it will retain a certain amount of fluid in the event of the rupture of, or leakage from, cylinder B. Furthermore, the upper edge of the socket serves as a reference point, indicating the safe level of fluid in the reservoir. *Patent No.* 746357. *Ford Motor Co. Ltd.*



No. 746357

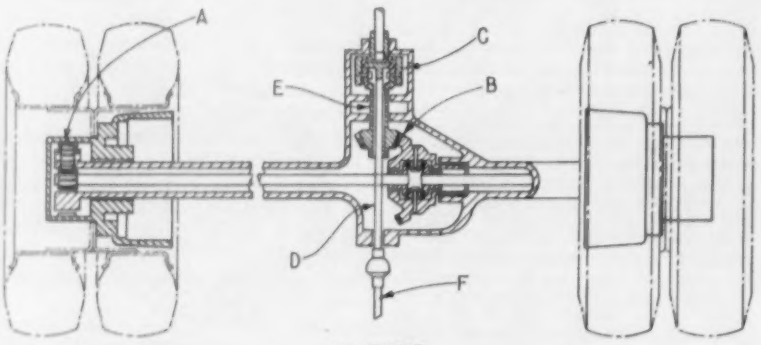
Tandem driving axles

In worm-driven axles, gearing appropriate for the transmission of the torques involved can be designed and the forward worm can be hollow to permit the incorporation of a third differential. Such an arrangement is not practicable should spiral bevel gearing be desired, as any rearward extension of the forward pinion shaft would intersect the axis of the forward axle. This objection does not arise with a hypoid bevel drive but in order to obtain the requisite reduction ratio the nose of the hypoid pinion would be too small to transmit the necessary torque and too small to accommodate the shaft of the third differential.

As shown in the illustration, the difficuty is overcome by the use of hub gearing A giving, in the example quoted, a reduction of 3.66 : 1. Hypoid bevel gearing B in the two axle drives has a ratio of 2.27 : 1, giving an overall reduction of 8.33 : 1. The drive to the bogie is through a third differential C, one shaft D of which passes through the hollow pinion E and is connected by cardan shaft F to the rear axle *Patent No.* 743027. *Kirkstall Forge Engineering Ltd.*

Fuel injector

Numerous advantages are claimed for this injector in which the nozzle is lifted off a stationary needle valve instead of the usual construction in which the needle is lifted off the nozzle. Combustion pressure assists in closing the valve and consequently the spring loading may be reduced. The valve opens when fuel pressure overcomes the sum of the spring pressure and the compression pressure of the engine. A higher fuel pressure is necessary to hold it open when combustion has commenced. This is desirable in a diesel engine where smooth combustion is obtained with an initial injection at low-pressure followed by higher pressure for the later portion of the injection.



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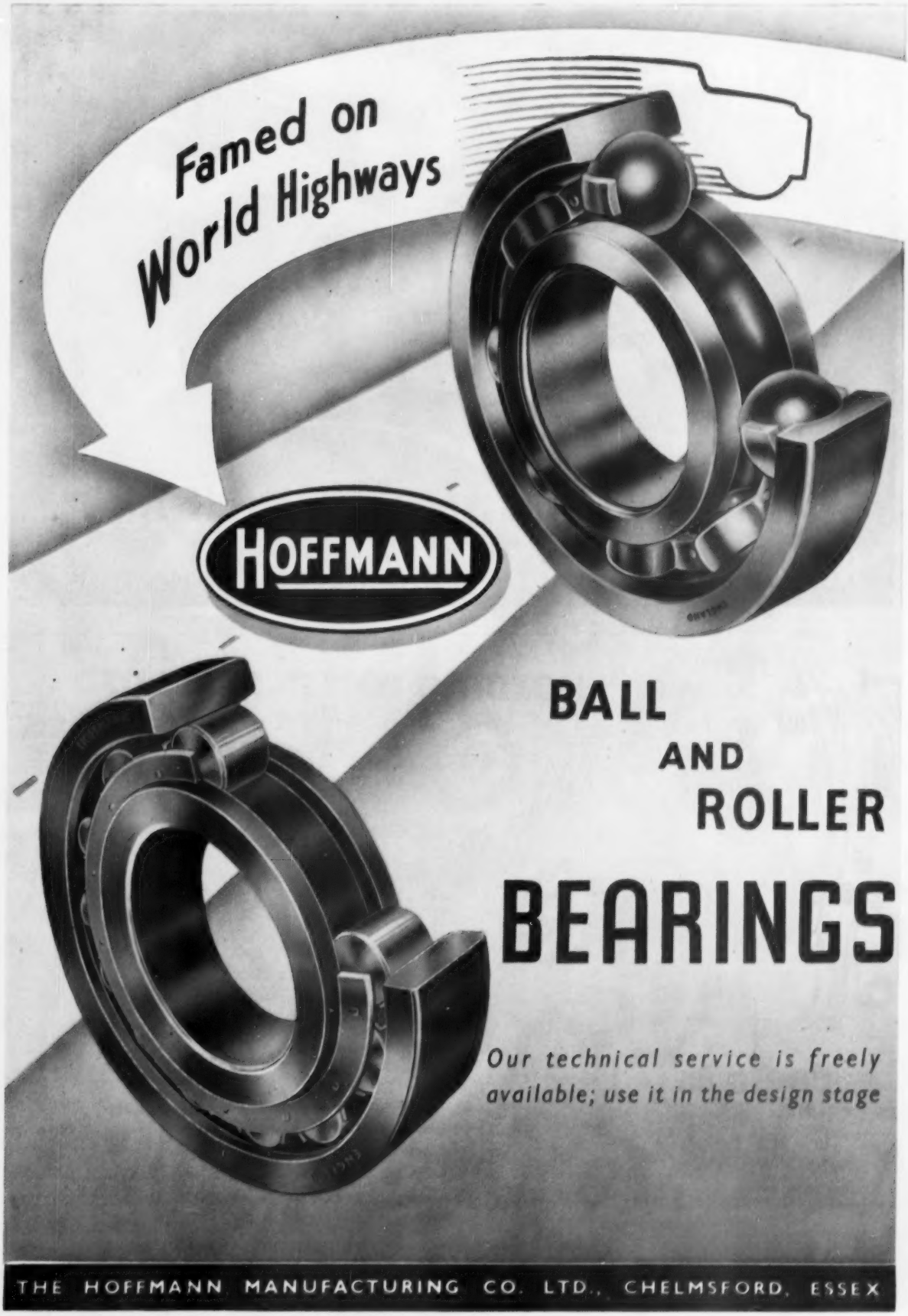
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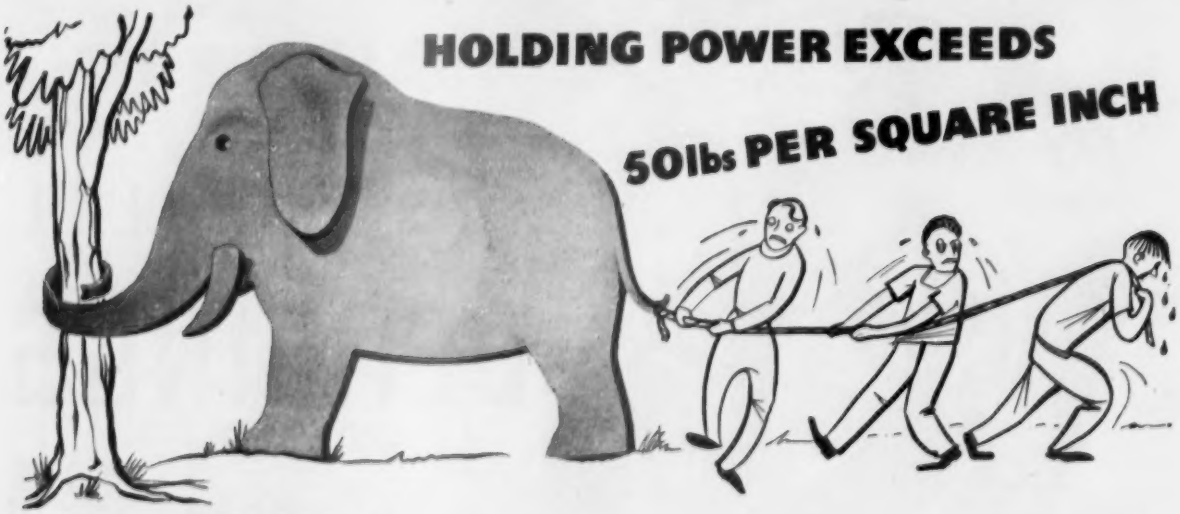

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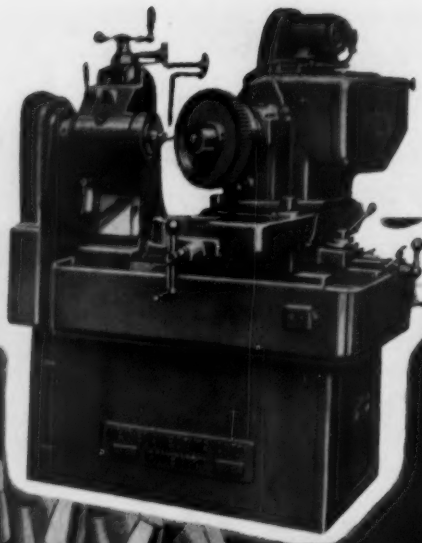

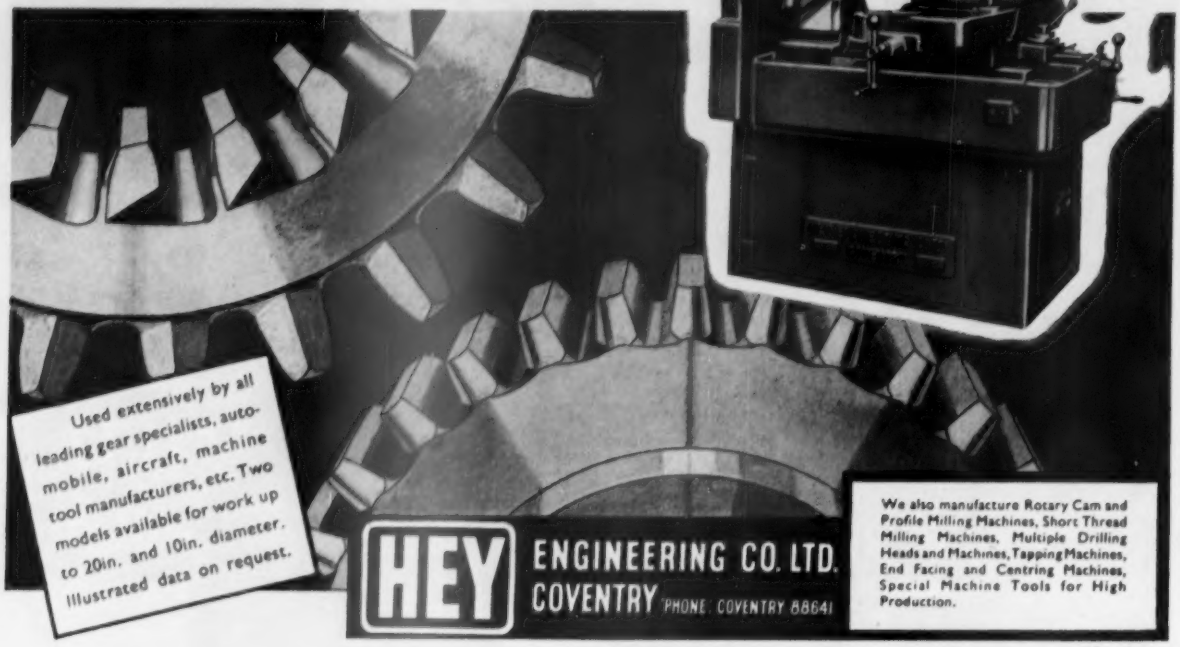

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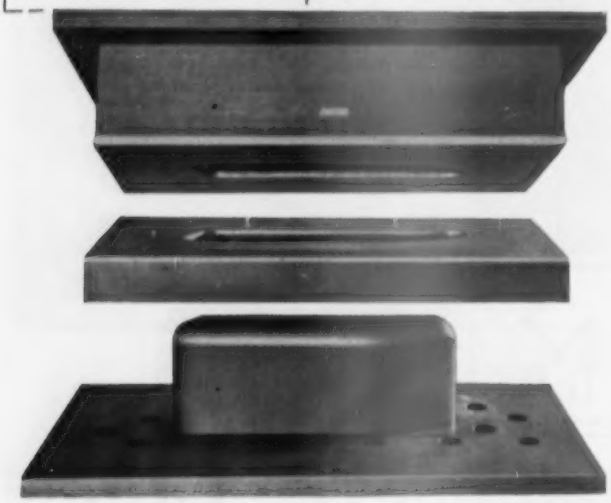

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an argument oft repeated

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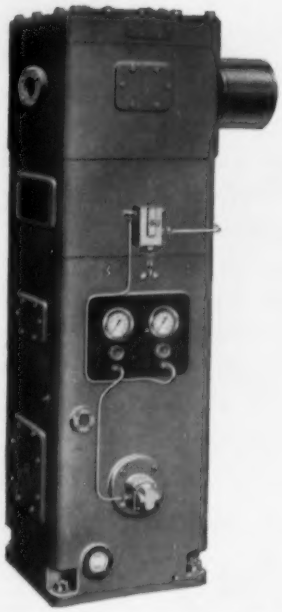
The illustration shows the Carbon Ring oil-free air compressor, symbol ODA2. This machine will deliver 200 c.f. of free air per minute to 30 lbs. p.s.i. running at 600 r.p.m.



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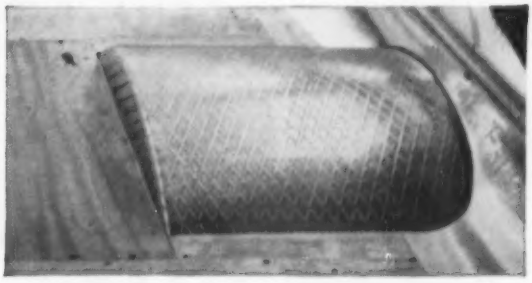
The illustration at left shows 'MG5' chequer plate used at a coach entrance.
Below. Wheel arches that will never rust! 'MG5' is ideal for jobs like this!



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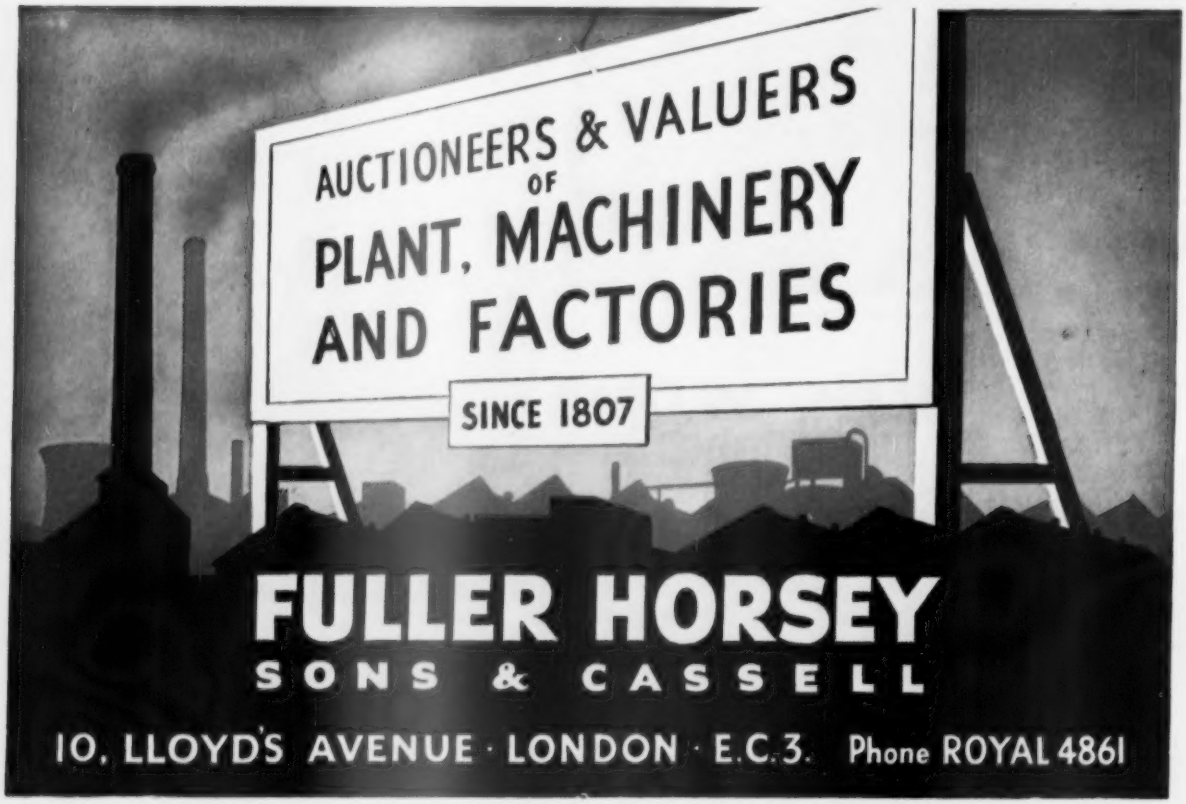

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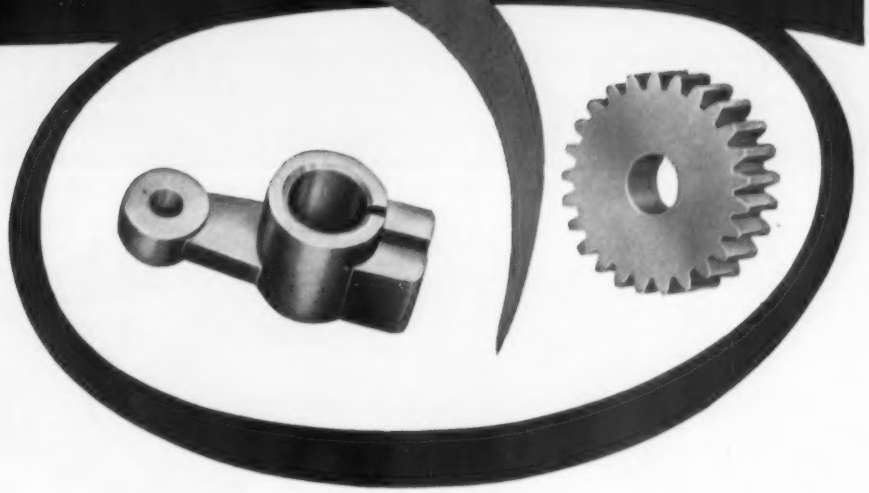

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The new works at Velindre, near Swansea, is built on a site of 48 acres. It began operating on July 10th and will eventually double the Company's production of cold-reduced tinplate.

Velindre is similar to the tinplate works at Trostre, which was completed in 1951. Trostre is already producing at a rate of more than 400,000 tons per annum. The equipment at Velindre will be of 95% British manufacture and will include electrolytic tinning lines capable of producing all weights up to 1 lb. and differential coatings. The combined resources of the Company's new plants will include five electrolytic tinning lines, 16 hot-dipped tinning pots, with automatic inspection and stacking and the widest range of size, substance and temper.

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As the production of this plant increases, the necessity for further tinplate imports—which in 1955 totalled 80,000 tons—will disappear.

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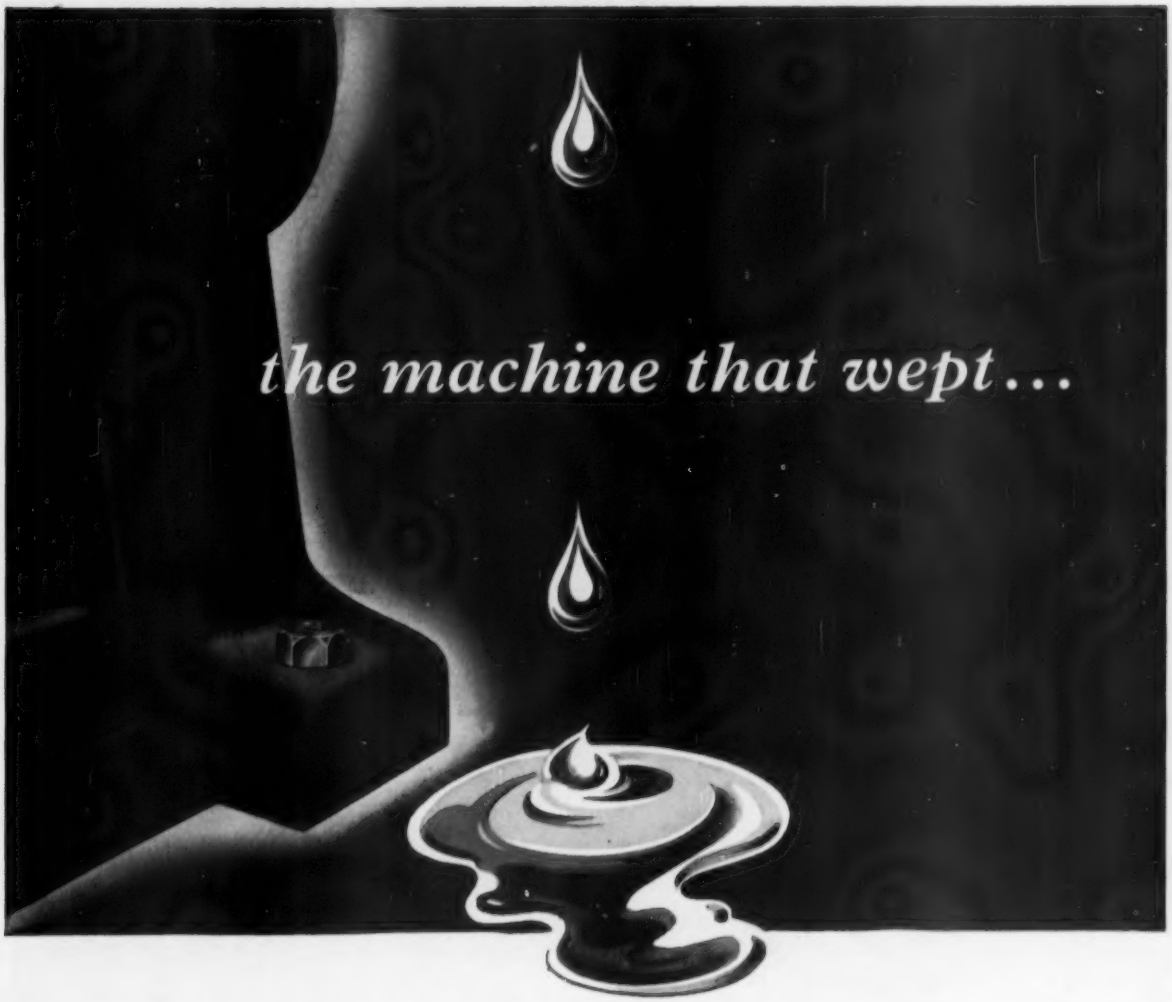

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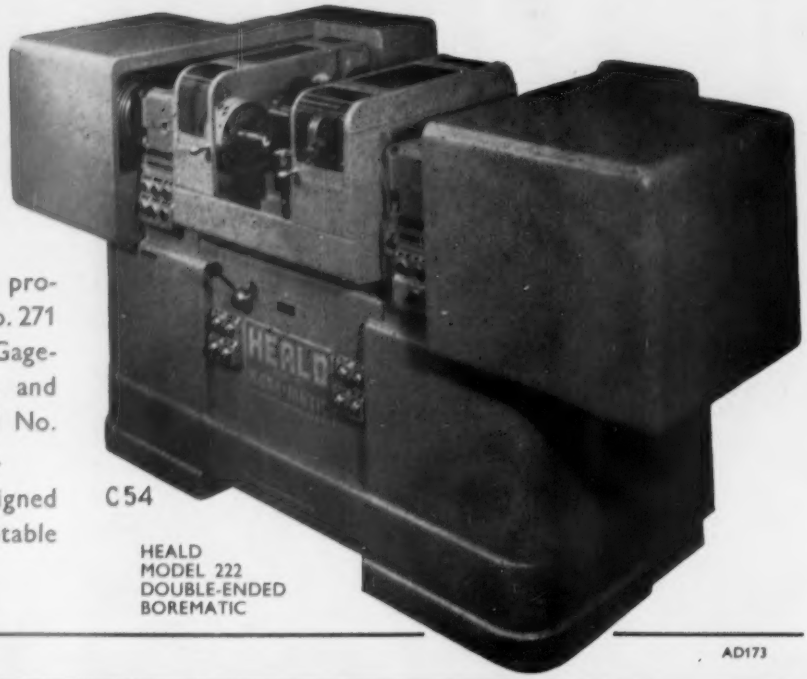

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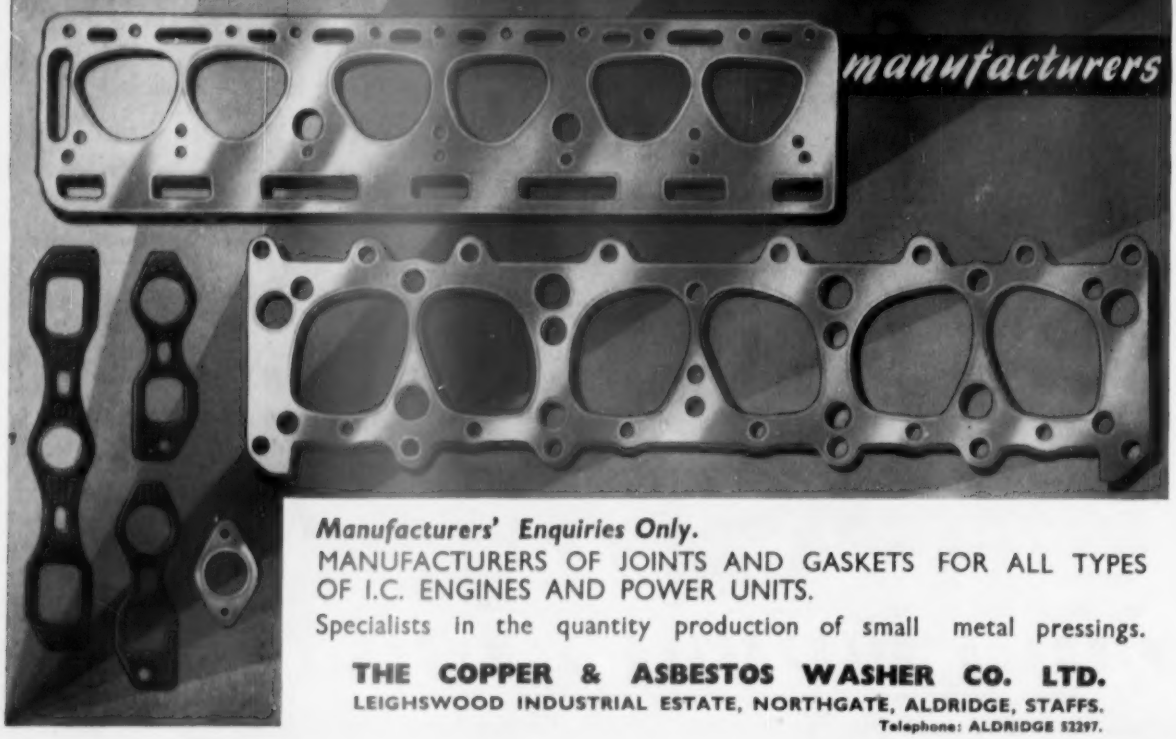

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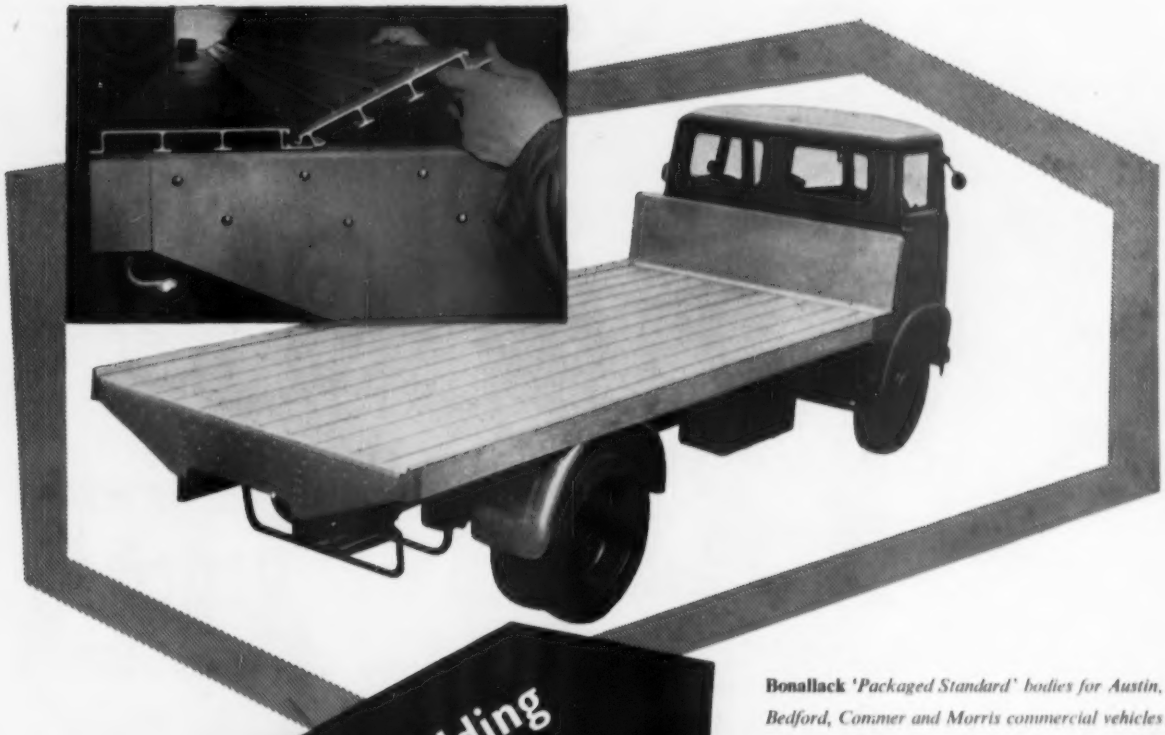

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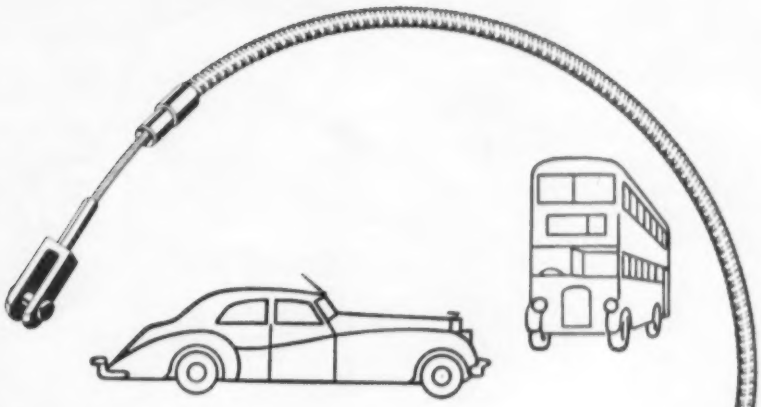

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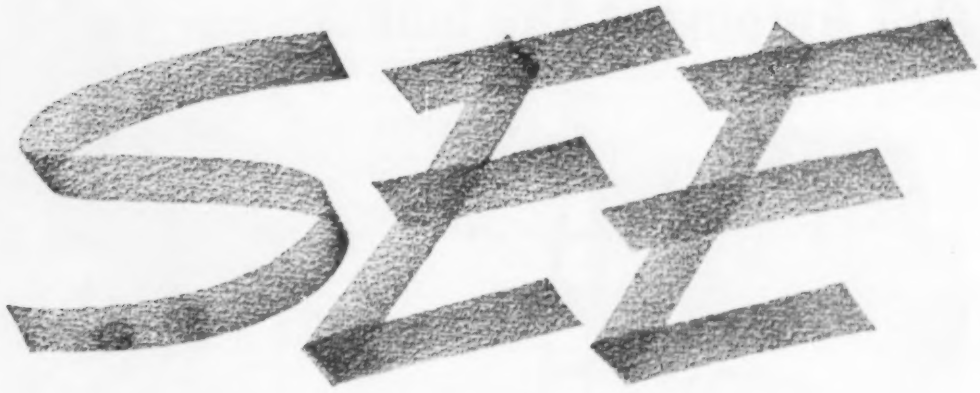

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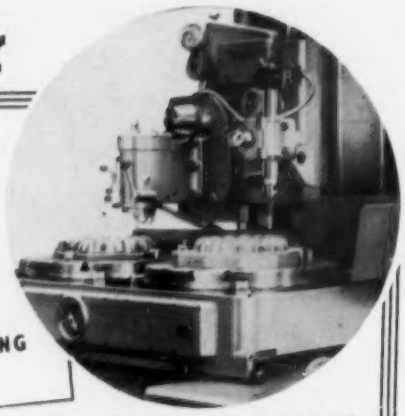

Illustration shows machine copying with vertical attachment fitted, and using the two rotary tables.

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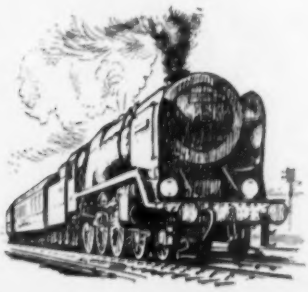

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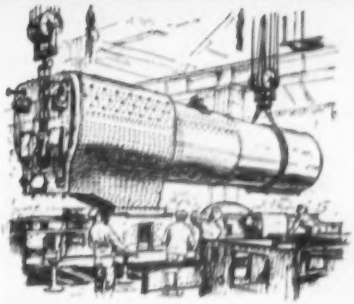

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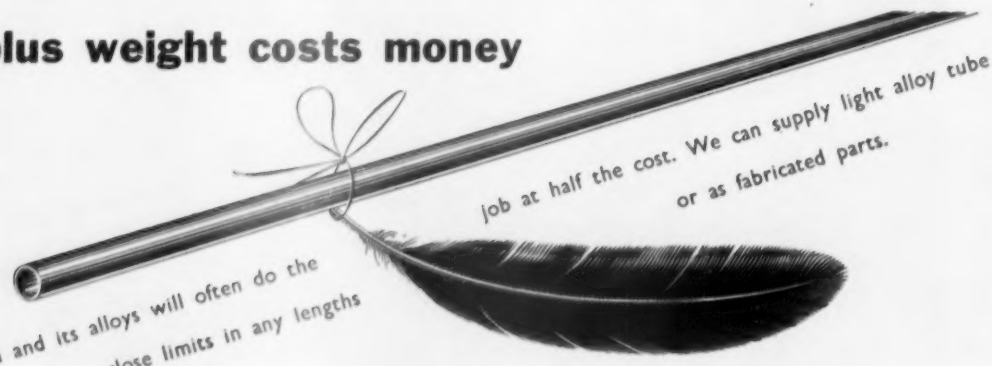

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No. 2 Soluble Quenching Oil outlasts other Quenching Oils, and based on ultimate cost, is the most economical to use.

We have a demonstration heat-treatment plant at our Birmingham works and our metallurgical staff will gladly advise on the selection and best application of our efficiency products.

Edgar Vaughan & Co. Ltd.
BIRMINGHAM · 4 · ENGLAND

Works and Depots at: Birmingham, Manchester, Liverpool, Southall (Middx.), Bristol, Glasgow

Lower oil consumption smoother running with DUAFLEX Oil Control Rings

This model 'O' Bedford, operated by Pease Transport Limited (Carriers for Tate & Lyle Limited), was fitted with Duaflex Rings when oil consumption was 1 pint in 88 miles. After fitting, the oil consumption fell to 1 pint in 405 miles and 56,000 miles were completed on this set of rings before oil consumption again rose to a prohibitive figure. This efficient Transport Organisation has equipped many other vehicles of different makes with Wellworthy Duaflex Rings and it is well satisfied with the general improvement in engine efficiency.

Duaflex Rings restore power and efficiency

More and more fleet owners are fitting Duaflex Rings in worn engines using an excessive amount of oil but not yet due for a major overhaul. Self-adjusting Duaflex Rings take up irregularities in the bore and form a gas-tight seal between pistons and cylinder walls, thus reducing oil consumption and restoring lost compression.

How Duaflex Rings work

1 The vertical sealing spring keeps the rails firmly in the ring groove, forming a perfect seal.

2 The expander maintains an even outward pressure on cylinder walls.

3 The rails 'wipe' oil from cylinder walls, avoid scraping and wear.

For best results fit Wellworthy Pistons with Matched Sets of Wellworthy Rings at the same time as fitting Duaflex Oil Control Rings.

WELLWORTHY

DUAFLEX

OIL CONTROL RINGS

'The Choice of the Expert'

REDUCE OIL CONSUMPTION
INCREASE COMPRESSION · DEFER RE-BORES

Write for literature to:—
WELLWORTHY LIMITED · LYMINGTON · HANTS

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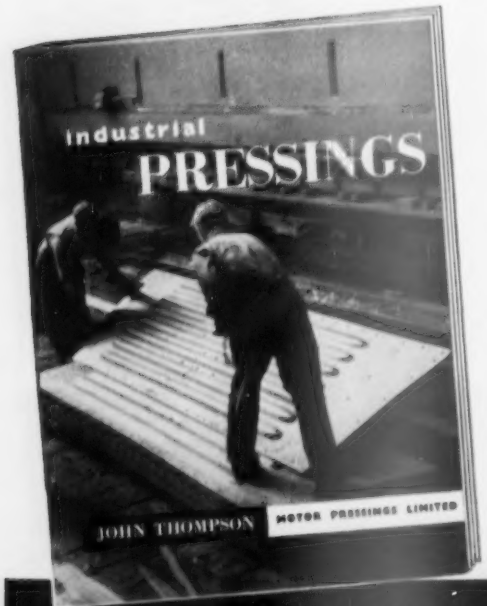

The necessary plant and long experience in heavy precision pressings are available in the John Thompson organisation for the production of all types of frames, components and fabrications for trucks, buses and heavy transport of all kinds. Preliminary discussion with experienced design engineers is freely offered. Write for illustrated brochure.

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JOHN THOMPSON **MOTOR PRESSINGS LIMITED**

When weight is a problem

Come to think of it, weight nearly always *is* a problem.

In almost every branch of industry—and particularly in transport—weight saved means greater all-round efficiency and economy.

That's where light, strong and durable 'Kynal' wrought aluminium alloys come in—enabling weight to be reduced without loss of strength. With the aid of the Technical Service and Development staff of I.C.I. Metals Division, engineers and designers are constantly finding new uses for 'Kynal' alloys.

May we help solve *your* weighty problems?

'KYNAL' AND 'KYNALCORE' wrought aluminium alloys are already extensively used in the following industries:

Aircraft: *ribs, spars, engine components, stressed skin covering, fittings, etc.*

Railways: *structural members, roofing, panelling, windows, luggage racks, etc.*

Road Transport: *structural members, floor planks and panelling, windows, tread strips, doors, small fittings, etc.*

Shipbuilding: *bridges, wheelhouses, outer funnels, lifeboats and davits, decks, skylights, stanchions, bulkheads, watertight doors, etc.*

Building: *roof coverings, side claddings, ventilators and windows, panelling, interior fittings, etc.*

'KYNAL' AND 'KYNALCORE WROUGHT ALUMINIUM ALLOYS

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M.414

IMPERIAL CHEMICAL INDUSTRIES LIMITED, LONDON, S.W.1

they go to Town*...

at Marshall Sons and Co Ltd

In the toolroom, an A.E.I. type Radial is used for accurate work, in this case drilling a jog plate, the holes having been plotted previously on a jig boring machine.

* FRED^K TOWN & SONS LTD HALIFAX · YORKS

ESTABLISHED 1903

T11

In the words of Shakespeare:

"This way for me"

King Henry VI (2nd part) Act III. Sc. II.

Queen Margaret, in choosing her way, knew she was going in the right direction. So will you if you choose the "Pop" Riveting System for the solving of your assembly problems.

Faster, more economical and enabling riveting to be done in one operation, by one operator, working from one side of the material only, the "Pop" Riveting System prevents distortion of the structure and enables rivets to be set in positions inaccessible by other methods.

A trial will convince you that the "Pop" Riveting System is the way for you.

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ONE OPERATOR OPERATION

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"POP" is a regd. Trade Mark of Geo. Tucker Eyelet Co. Ltd.

THE MODERN SYSTEM OF RIVETING

Perfect Power Transmission

Precision built Hardy Spicer propeller shafts and universal joints convert the power smoothly and silently into swift and unrestricted movement. And today most British vehicles are fitted with Hardy Spicer shafts and joints. Replacement parts are available throughout the world.

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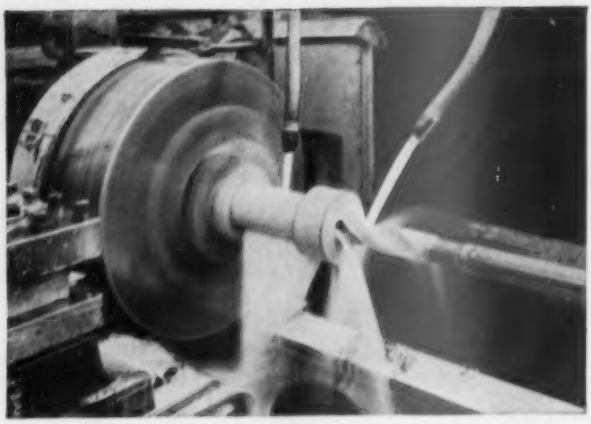
NEEDLE BEARING
UNIVERSAL JOINTS & PROPELLER SHAFTS

HARDY SPICER LIMITED · (*A Birfield Company*) · BIRCH ROAD · WITTON · BIRMINGHAM 6



More output per re-grind —thanks to F.M.!

During a broaching operation on Rex 448 steel, production of fir tree forms on aircraft turbojet discs was stepped up from 62 to 880 forms per re-grind of the broach by the introduction of Swift R.10 Broaching Oil—an increase of 1,320 per cent.

Achieving a standard of finish never before accomplished by any previous broaching oil, Swift R.10 is only one of 46 high-grade cutting fluids that can help you to attain better results from your machine tools.

Here are two more interesting speciality oils from the Fletcher Miller range:—

Swiftcol Heavy Duty Soluble Oil

This soluble medium often replaces straight oil on specialised cutting operations such as broaching, open-type threading and non-intricate automatic lathe work. The emulsion is semi-clear, concentrations ranging from 1:3 to 1:15.

Pale Swiftex Straight Oil

In distinct contrast, here is a non-sulphurised oil that gets maximum results when cutting high tensile steels and non-ferrous metals of low machinability rating. It is recommended for use with jig borers, gear generators and cam lathes.

Send for the latest edition of our Publication SP.173, "Cutting Fluids," on your trade card or heading, please.

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cutting fluids

— YOUR PARTNERS IN PRODUCTION

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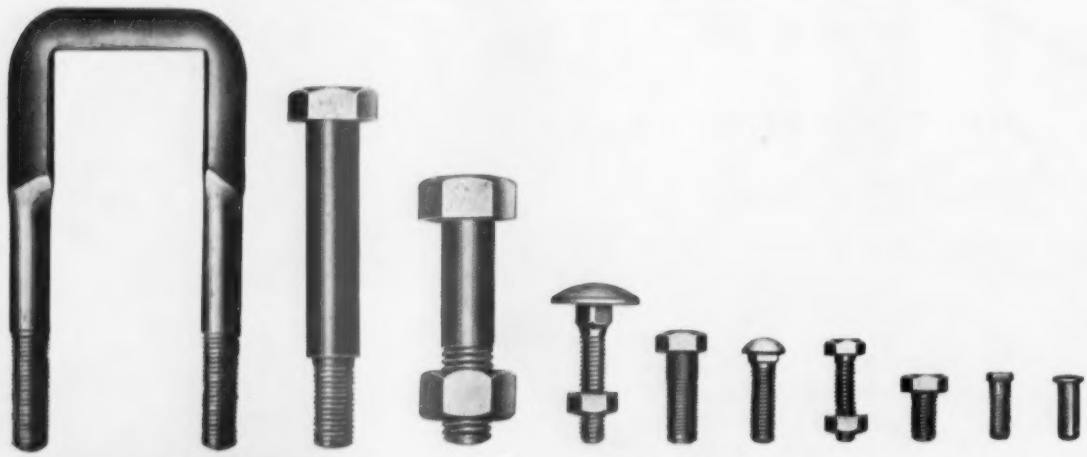


WILEY

Whatever your fastening problem, Wiley can make a bolt for it—and a nut too—ask them !

make a bolt for it!

JAMES WILEY & SONS LIMITED, DARLASTON, STAFFS. *Telephone : James Bridge 2692.*

M-W.61

TOUGH AND RIGID YET EASY TO MACHINE

Gloucester Blackheart Malleable castings are widely used throughout the automobile, railway and farm machinery industries.

The new elevator-type electric furnace anneals in 48 hours! – instead of the usual 7 days. It also allows greater control of the component during annealing, resulting in uniformity of metal, greater strength and resistance to impact, and a smoother,

GLOUCESTER

malleable & grey iron castings

finer skin. Machining time is therefore reduced to a minimum. Every heat is under constant laboratory control throughout the day and castings are rigidly inspected at every stage of production.

Gloucester technicians will gladly advise on the design of malleable or grey iron castings. Why not call them in at the drawing board stage?

Typical Gloucester Malleable specifications are:

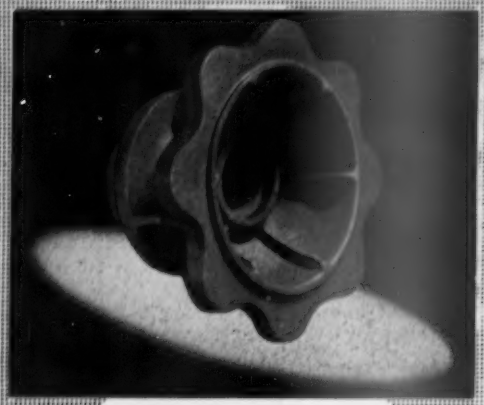
Gloucester Blackheart Malleable		*Gloucester Lamellar Pearlitic Malleable*	
Elongation	18%	Elongation	5%
Yield Point	15 tons	Yield Point	22 tons
Tensile strength	25 tons p.s.i.	Tensile strength	35 tons p.s.i.

(right) Differential cage for motor vehicle in Malleable Iron.
(left) Brake Shoe in Malleable Iron.

GLOUCESTER FOUNDRY LTD., EMLYN WORKS, GLOUCESTER
(A subsidiary of the Gloucester Railway Carriage & Wagon Co. Ltd.)

Telegrams: 'Pulleys' Gloucester
Telephone: Gloucester 23041

From first thoughts to the FINISHED CASTING



For your production under way and for products still under discussion, we can be very useful. Bring us in at the design stage and we can probably suggest improvements or economies; obviate snags *before* they hold up production. We believe in co-operation. Not only does it help you—but it helps us to do a better job for you. And when it comes to capacity, we can probably meet your entire programme—and arrange deliveries to suit your requirements.

Under strict laboratory control at every stage of production, our Blackheart Malleable Iron Castings are made to such close limit accuracy that castings are invariably ready for immediate use on customers' machining jigs. This makes an immediate saving in material, time and labour.

BLACKHEART *Malleable* IRON CASTINGS

SHOTTON BROS·LTD

OLDBURY, NEAR BIRMINGHAM

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NOW SATISFY YOUR DEMAND FOR **PERFECTION**

Use the *right* masking tape for every task. Permacel masking tape is used by most leading manufacturers. Here are examples of two tapes — and their uses.

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The Use: Protecting machinery during manufacture.

The Tape: Permacel Self-Adhesive No. 30 Unbleached Cloth Masking Tape, made of unbleached cotton cloth, will stick instantly and firmly to any surface, yet comes away cleanly and easily. It is unaffected by many of the chemicals normally used in paints and dopes.

Avery-Hardoll Ltd.

The Use: Protection during the spraying of Petrol Pumps.

The Tape: Permacel Self-Adhesive Crepe Paper Masking Tape is a specially prepared tape for spray painting. It is highly flexible, readily conforms to uneven surfaces and goes around corners, evenly. It resists moisture and solvents in paints, varnishes, enamels and lacquers.

A GUIDE TO PERFECTION: *The Permacel brochure helps guide you through the intricacies of packaging tapes and subsidiary equipment. Over 400 examples classified. Send for your FREE copy to-day.*

PERMACEL TRADE MARK

INDUSTRIAL MASKING TAPES

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Johnson & Johnson (GT. BRITAIN) LTD., SLOUGH, BUCKS.

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road to-day
owes
some
part
to Wilmot Breeden
STAND No. 328 MOTOR SHOW

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An Aberdeen-Angus was first shipped to the Argentine in 1876—fourteen years after the Bank of London & South America had opened its doors there. Since then, commerce with Latin America has increased enormously, and with it have grown the influence and interests of the Bank. Today, the Bank holds a unique position as the *only* British bank in Latin America.

It has branches in most of the important trading centres of that vast continent.

Up-to-date reports on local markets, trading regulations and general economic conditions are received regularly from Branches overseas and are summarised in the Bank's *Fortnightly Review*, which is distributed to customers and made available to those who have business in Latin America, Portugal or Spain.

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& SOUTH AMERICA LIMITED

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Famous Quotations

"There is nothing more requisite in business than despatch"

Joseph ADDISON *(1672-1719)*

"It has long been an axiom of mine that the little things are infinitely the most important."

Sir Arthur Conan DOYLE *(1859-1930)*

"Genius is one per cent inspiration and ninety-nine per cent perspiration."

Thomas Alva EDISON *(1847-1931)*

"The best thing in all this world that can befall a man is to be born lucky."

Samuel CLEMENS (Mark Twain) *(1835-1910)*

"The glory of a workman, still more of a master-workman, that he does his work well, ought to be his most precious possession; like the 'honour of a soldier', dearer to him than life."

Thomas CARLYLE *(1795-1881)*

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ALUMINIUM & NON-FERROUS CASTINGS
LET SANDWELL QUOTE

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Specialists in the production of the finest quality Sand Castings, Gravity Die Castings and Pressure Die Castings in all Light Alloys and Non-Ferrous Metals by the most modern methods.

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BANK STREET FOUNDRY, WEST BROMWICH
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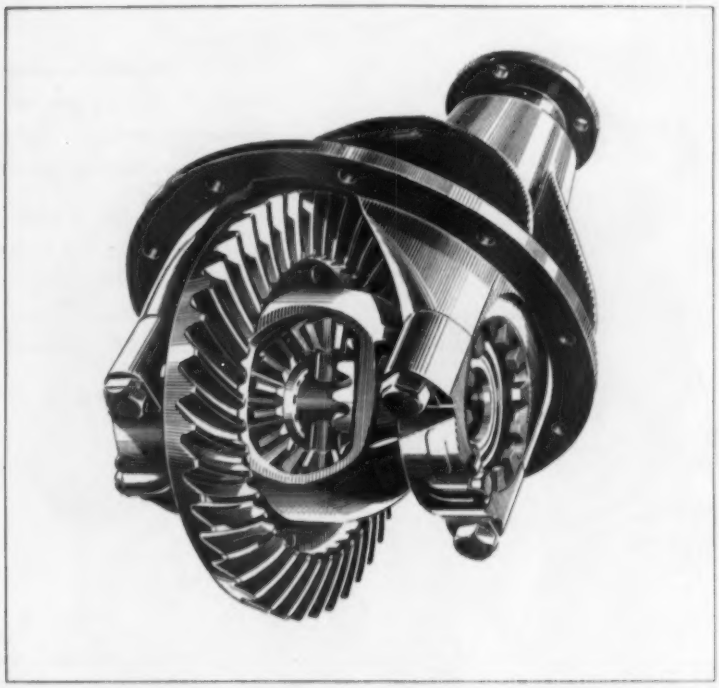


ENV

for Gears

Specialists in spiral bevel and hypoid gears for rear axles, differential components and assemblies, automotive gear boxes and transmission gears and components. Camshafts and timing gears.

Stand No. 364

E.N.V. ENGINEERING COMPANY LIMITED *HYTHE RD., WILLESDEN, LONDON N.W.10*

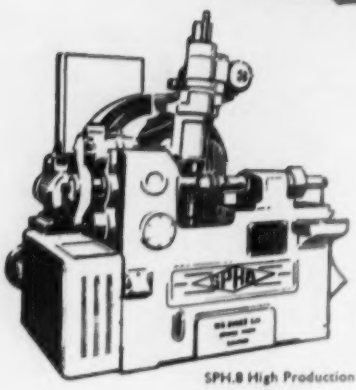
AP. 265-57.

in 60 seconds

the time it takes to read this advertisement . . . .

Ideal for automobile engineering, the SPH.8 is a new machine for hobbing spur and helical gears to finished or pre-shave standards. High speed operation ensures that gear cutting delays are entirely eliminated.

The gear illustrated is 3·188" p.c.d. and ¾" face width, with 29 teeth of 11 N.D.P., 30·42° helix angle and 20° pressure angle. The material is EN 18C, Manganese Chromium oil hardening steel, and a floor-to-floor cutting time of only 60 seconds is consistently obtained.



SPH.8 High Production Hobbing Machine

the SPH8 will cut this gear

SYKES

W. E. SYKES LTD.

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Britain's Roman Highways

ERMINE STREET

When the Romans invaded Britain they were faced with the problem of creating suitable highways which would give their legions access to the ends of the country. Of the many they built the most important and famous were the Four Ways; Watling Street, The Fosse, Ermine Street, and Icknield Street.

Ermine Street, the third of the Four Ways, was a direct road from London to Lincoln and on to Wintringham on the Humber. By means of a ferry it made communication with Brough and York. North of Lincoln another road diverged from it Westwards to Doncaster. Near London the Enfield Highway is part of the old road.

In Lincolnshire, Ermine Street is known as a 'ramper' or 'The Old Street.' Like Watling Street and the Fosse Way, Ermine Street appears to have been paved from end to end.

Today the historic highway has become part of the main road transport system of the country. This system is about to be modernized so that the transport of the country may continue to flow evenly.

Richard Berry have been actively concerned in improving the passage of vehicles over the roads of Britain and the World for over a hundred years. Today the tradition of their skilled craftsmen is being kept alive by the products of their new road spring factory.

R B TRADE MARK

write for your copy of
"LEAF SPRING DESIGN"
by Alan Hodgson

CARRY ON BERRY

RICHARD BERRY & SON, WEST BROMWICH, STAFFS.

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ONE OF THE BROCKHOUSE COMPANIES

We like to point out to people that practically every British vehicle on the road has some Rubery Owen component or other in its construction. And we know our friends, the manufacturers, are happy for us to stake our claim in one of Britain's Greatest Industries.

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Wins on points

How can we be so sure of the superiority of TURQUOISE pencils? Because every process in their manufacture is tested and proved in the EAGLE testing rooms to give long and satisfactory service. Use TURQUOISE always.

THIS REFLECTOMETER checks blackness for grading.

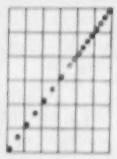
Test charts of pencil shadings, held under the electric eye of the Reflectometer, register on the dial as exact percentages between black and white. Opacity to a fraction of one per cent is recorded.

THIS GIANT PENDULUM proves the smoothness of a lead.

As the point of the lead presses on to the paper the friction slows or stops the pendulum. The smoother the lead, the longer the pendulum swings, thus determining the relative writing smoothness of different lead formulae. The 540-lb. bob swings freely, making 40-50 oscillations from a single impulse before coming to rest.

THIS PRESSURE SCALE breaks points so that you won't.

The pencil is pressed down on to the plate at the normal writing angle. The dial indicates the exact pressure required to break the point.



THIS CHART shows difference in hardness.

This chart illustrates the exact difference in hardness between each degree. Because the technical user requires great accuracy, they are spaced twice as closely from F to 9H.

THIS METER tests length of line.

On this revolving drum the pencil makes a continuous line, which is measured by multiplying the circumference of the drum by the number of revolutions, and thus tests the durability of EAGLE pencil leads.

'CHEMI-SEALED' TURQUOISE DRAWING HB

'CHEMI-SEALED' TURQUOISE DRAWING HB

EAGLE

'CHEMI-SEALED' *Super-bonded*

TURQUOISE

DRAWING PENCILS

with **100% 'ELECTRONIC'** *Graphite*

T.13

EAGLE PENCIL COMPANY, ASHLEY ROAD, TOTTENHAM, N.17

quality in special purpose steel

DuneIt

ALLOY AND SPECIAL CARBON STEELS

Black rolled, bright drawn or smooth ground, in heat-treated or unheated conditions. Free cutting steels, heat-resisting steels, die steels, shear blade steels, high-speed tool steels, stainless steels, valve steels, hollow steel bars.

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Telephone: 41121 (5 lines). *Telegrams*: "Blooms, Sheffield, 9".
London Office: 54 Victoria St., S.W.1.
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If it can be Riveted . . .

HERE IS HOW YOU CAN SPEED UP THE JOB—AND CUT COSTS!

If you produce riveted assemblies where speed in operation and a neat appearance are as important as a properly formed rivet head—it will pay you to investigate T.T. Rivet Spinning machines. The wide range of machines available are suitable for cold riveting in all sizes up to $\frac{3}{4}$" diam. mild steel. It includes bench and floor models, air and foot operated and "Duplex" spinners for heading both ends of pins for hinges, chains, etc. Write for full details and latest literature.

TT RIVET SPINNING MACHINE

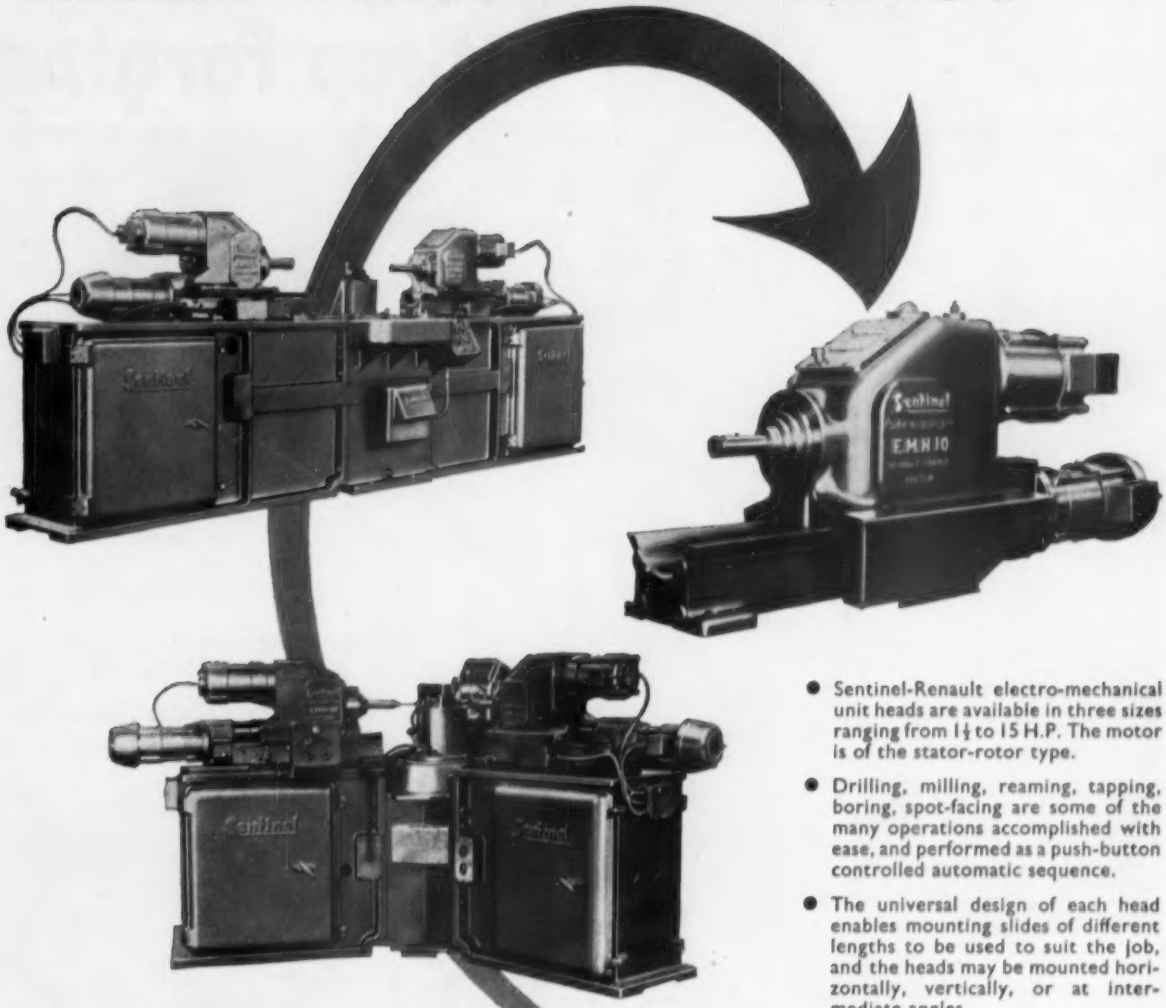
R.S.S. VMD. $\frac{5}{16}$in. capacity. Foot Operated Model.

TURNER
MACHINE TOOLS LTD
63-68, PRINCIP ST., BIRMINGHAM, 4

Ask for a demonstration. Full technical advisory service available. Samples of articles would be appreciated.

Tel: ASTon Cross 2244

AHEAD WITH DELIVERIES!



- Sentinel-Renault electro-mechanical unit heads are available in three sizes ranging from 1½ to 15 H.P. The motor is of the stator-rotor type.
- Drilling, milling, reaming, tapping, boring, spot-facing are some of the many operations accomplished with ease, and performed as a push-button controlled automatic sequence.
- The universal design of each head enables mounting slides of different lengths to be used to suit the job, and the heads may be mounted horizontally, vertically, or at intermediate angles.

Ahead with deliveries—and a head capable of a range of almost limitless operations. The use of standard constructional parts such as columns and rotary tables, leaves fixtures and tooling only to be made to your specification. For this reason Sentinel can meet your special machine tool requirements very quickly. Sentinel machines ensure economy for the future, since, in the event of changed production schedules, many of the standard parts may be incorporated in your new machine.

Modern manufacturing methods at our Shrewsbury factory enable us to offer EX STOCK deliveries of units, to meet the demands of mass producers, as well as those of progressively minded batch-production concerns.

Sentinel

SENTINEL (SHREWSBURY) LIMITED SHREWSBURY

TELEPHONE SHREWSBURY 2011 TELEGRAMS SENTNOLL SHREWSBURY

A METAL INDUSTRIES GROUP COMPANY

SS/22

FRY'S
Solder Quiz

Here's your chance to see how much you know about modern soldering techniques! Please try your luck with our two minutes quiz.

This "action shot" was taken in a famous manufacturer's sheet-metal shop.

1. The unit being dipped is . . . (a) A condenser? (b) A water-heater body? (c) A margarine emulsifying chamber?
2. The solder bath under the unit contains . . . (a) Fry's No. 3 DIPPING ALLOY? (b) A special FRY'S dipping solder? (c) FRY'S Pure Tin?
3. The bath in the foreground contains . . . (a) Hot FRYSOL flux? (b) De-greasing fluid? (c) Cold FRYSOL flux?

WHAT DO YOU KNOW?
1. (b) Made by ASCOT—the water heater manufacturers.
2. (b) Produced especially for ASCOT, FRY's lead-tin dipping alloy.
3. (a) Used hot, FRYSOL penetrates, cleans and fluxes *quickly* and reduces spilling, allowing faster immersion of water heater body.

WHAT DO WE KNOW?
Why not send us your solder problems? Our Research Department will be pleased to offer you their free advice.

Presented for your interest by :—
FRY'S Metal Foundries Ltd., Tandem Works, Merton Abbey, London, S. W. 19.
Telephone MITcham 4023.
And at MANCHESTER · GLASGOW · BRISTOL DUBLIN
M.R.P.56

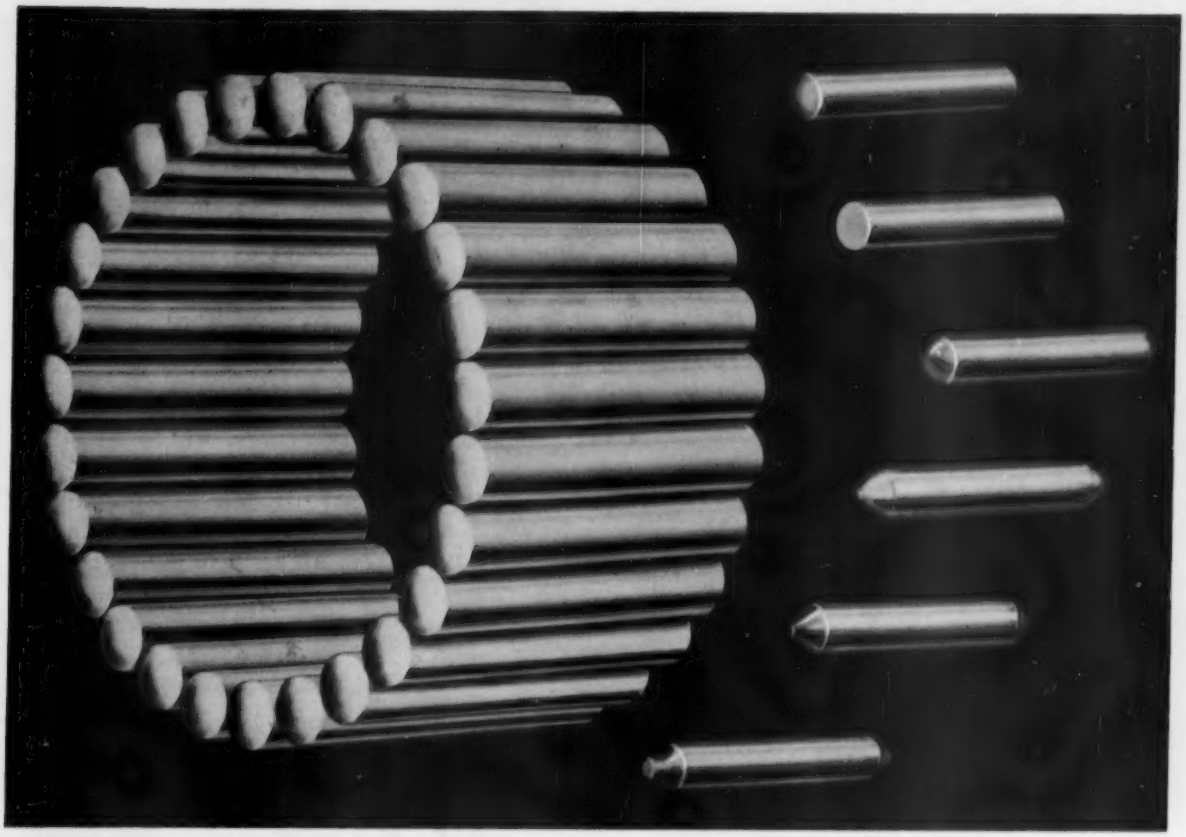
Good! a MITRE Drop Forging

We invite you to send for our sales folder which explains why MITRE precision Drop Forgings help to step up quality and production and reduce costs.

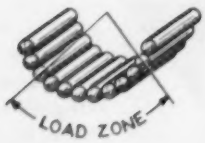
A. J. VAUGHAN & CO (MITRE WORKS) LTD
WOLVERHAMPTON ROAD WILLENHALL, Staffs. Phone 486/7

TORRINGTON NEEDLE ROLLERS

for the ultimate in load capacity at low cost

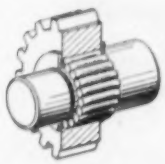


High radial capacity in minimum space is provided by full complement of small-diameter rollers. Load zone contains maximum number of contact lines.



High carbon chrome steel is carefully hardened, ground and lapped to make each Needle Roller a precision part for long-life performance.

Close tolerances on Needle Rollers are assured by stringent quality controls. Standard OD tolerance is .0002". Tolerance on length dimension depends on end formation.



Mounted without races between hardened shaft and hardened housing (RC-60 recommended), Needle Rollers allow largest possible shaft diameters.

Fine finish contributes to efficient anti-friction performance. Needle Rollers are usually polished to 4-6 rms finish.

Good operating results with Needle Rollers require careful design of mating parts furnished by the user.

Our Bearings Division with broad experience in applications of Needle Rollers will be glad to give you technical advice on your needs.

THE TORRINGTON COMPANY LTD.
Bearings Division:
Torrington Avenue, Coventry
London Office: 7-10 Eldon Street, EC2
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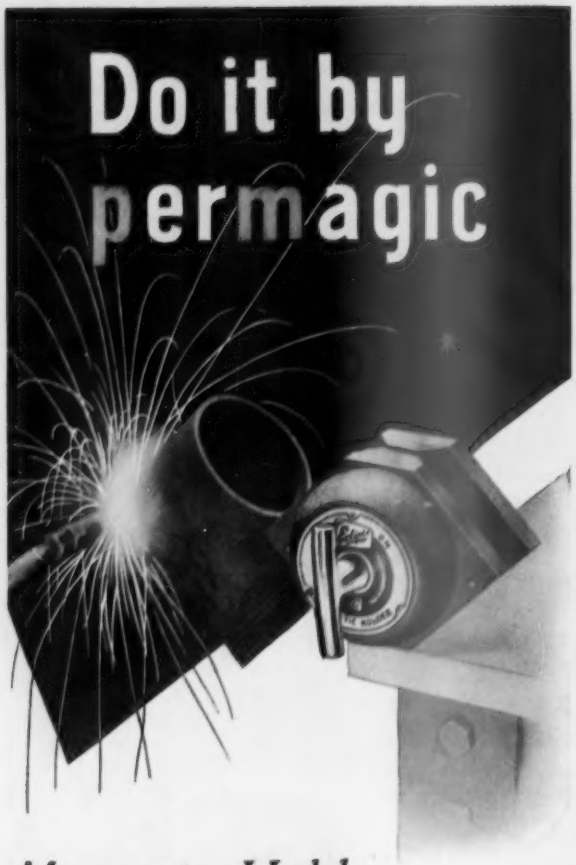

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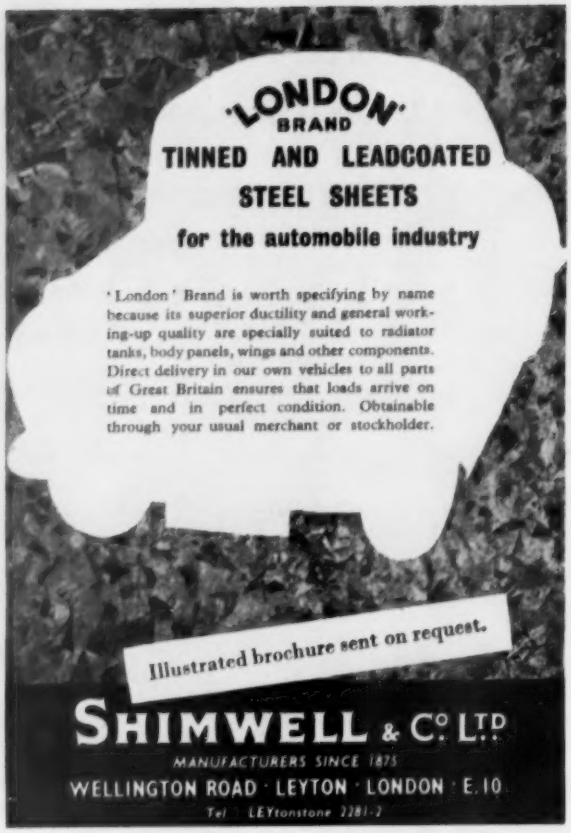
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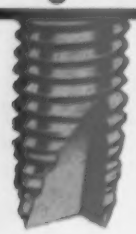

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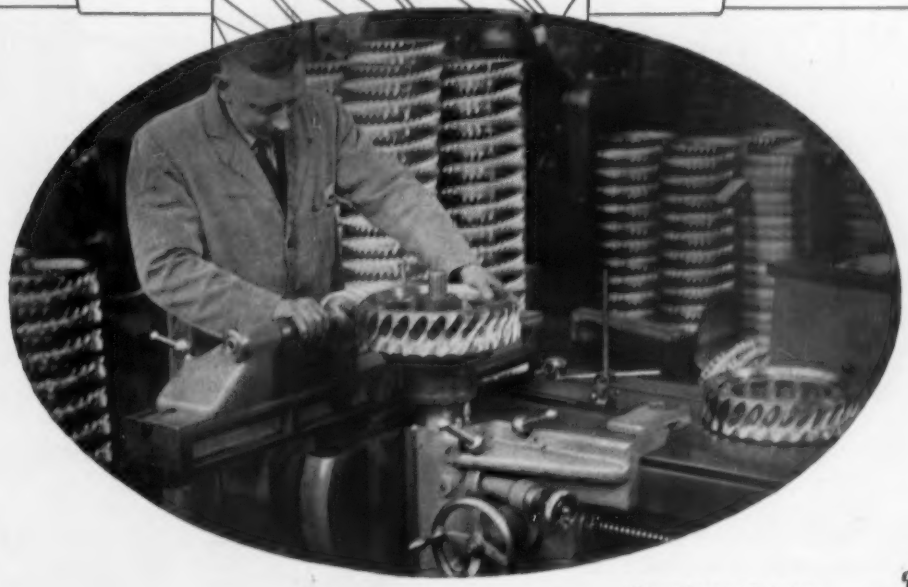

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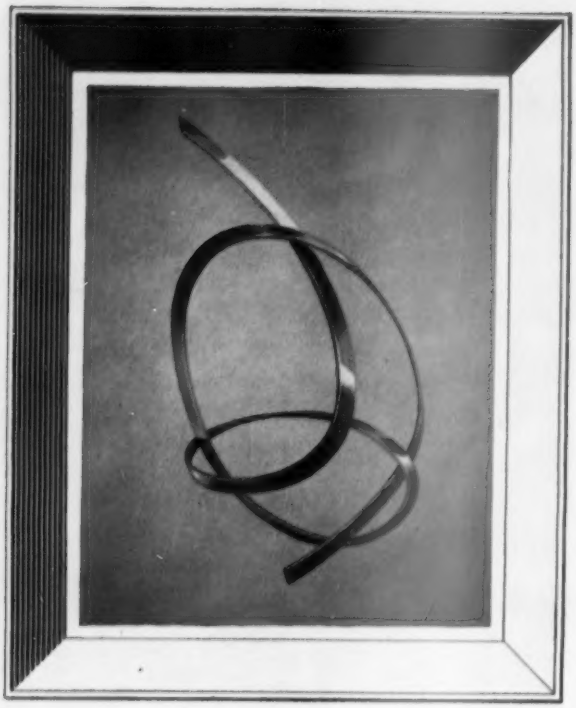

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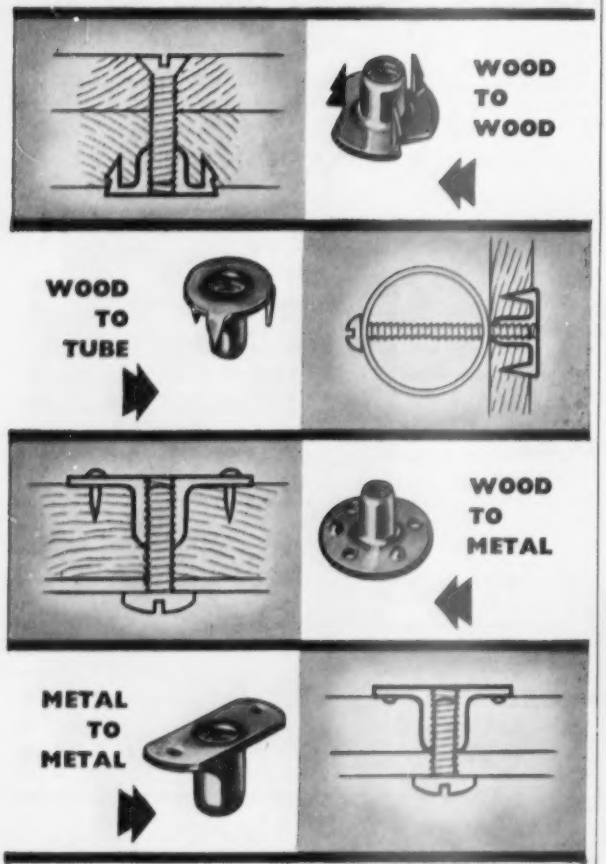

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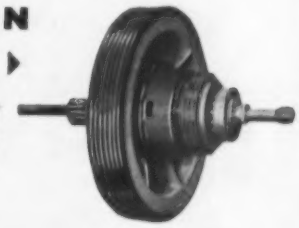

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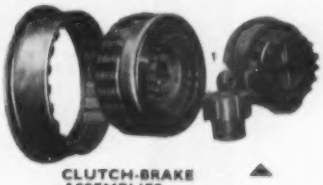

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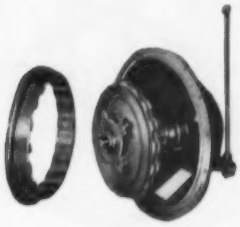
The illustration shows a pneumatic Clutch-brake Assembly supplied for a 500 ton Forging machine. Duty 1,100 h.p. at 42 r.p.m. frequency 3-4 strokes per minute.



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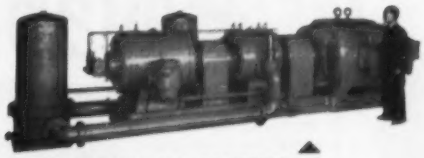

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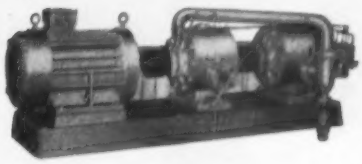

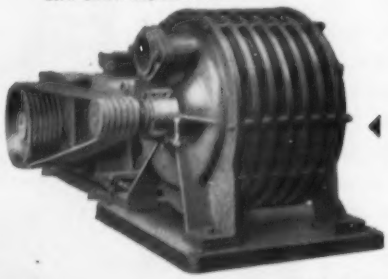

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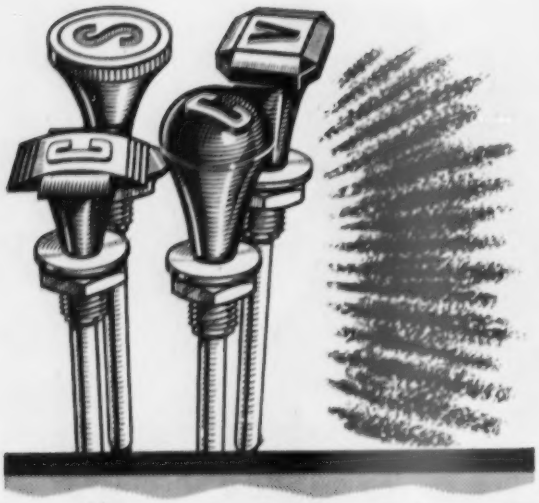

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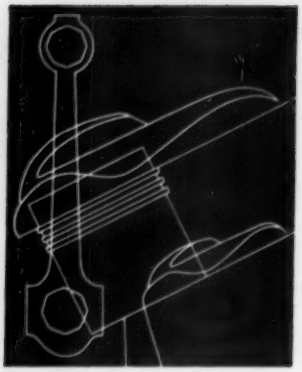

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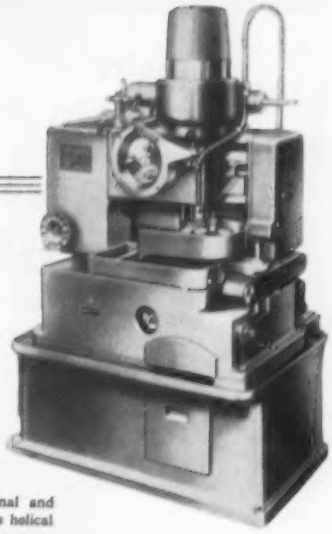

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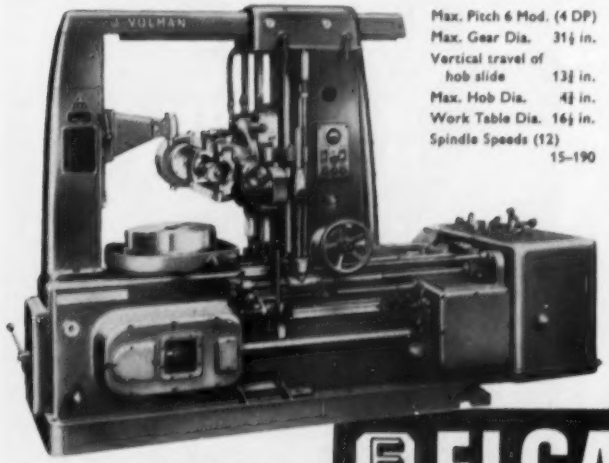

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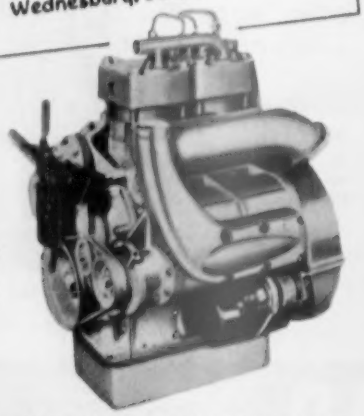

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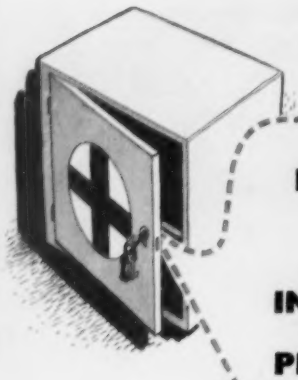

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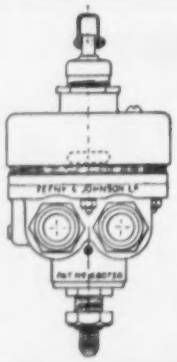

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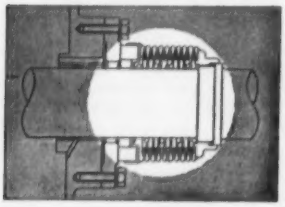

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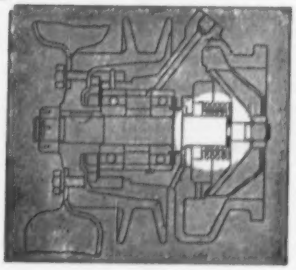

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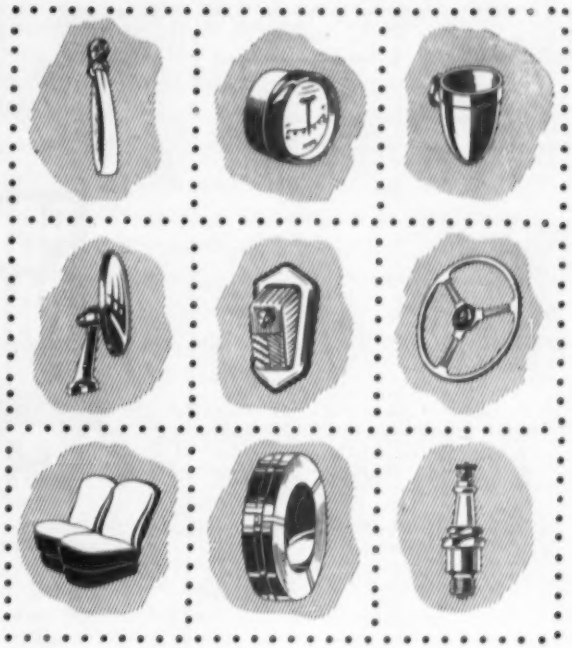

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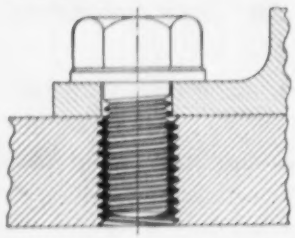

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Printed in Great Britain for the Publishers, ILIFFE & SONS LTD., Dorset House, Stamford Street, London, S.E.1, by James Cond Ltd., Charlotte Street, Birmingham 3. "Automobile Engineer" can be obtained abroad from the following: AUSTRALIA & NEW ZEALAND: Gordon & Gotch Ltd. INDIA: A. H. Wheeler & Co. CANADA: The Wm. Dawson Subscription Service Ltd.; Gordon & Gotch Ltd. SOUTH AFRICA: Central News Agency Ltd.; Wm. Dawson & Sons (S.A.) Ltd. UNITED STATES: The International News Co. Entered as Second Class Matter at the New York, U.S.A., Post Offices.

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